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MERDC/OSU HYDRAULIC SYSTEM
RELIABILITY PROGRAM

PREPARED BY PERSONNEL OF
FLUID POWER RESEARCH CENTER
OKLAHOMA STATE UNIVERSITY
STILLWATER, OKLAHOMA

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PREPARED FOR
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PREPARED FOR

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AND DEVELOPMENT COMMAND

Fort Belvoir, Virginia 22060

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– FOREWORD –

This report was prepared by the staff of the Fluid Power Research Center of the School of Mechanical & Aerospace Engineering, Oklahoma State University of Agriculture and Applied Sciences. The study was initiated by the Mobility Equipment Research and Development Command, Fort Belvoir, Virginia. Authorization for the study reported herein was granted under Contract No. DAAK02-75-C-0137. The time period covered by this report is from 1 February 1975 to 31 January 1976.

The Contracting Officer's Representative was Mr. Hansel Y. Smith, and Mr. John M. Karhnak served as the Contracting Officer's Technical Representative. In addition, Mr. Paul Hopler has effectively represented the Contracting Officer both technically and administratively through various phases of this contract. The active participation of Messrs. Smith, Karhnak, and Hopler during critical phases of work contributed significantly to the overall success of the program.

The studies represented by this report were conducted under the general guidance of Dr. E. C. Fitch, Program Director. The details of each study are presented in a self-contained section of this report. The titles of the various sections together with their respective Project Managers are listed below:

SECTION I.	**STRUCTURAL ANALYSIS OF CYLINDERS – S.K.R. Iyengar**
SECTION II.	**HYDRAULIC NOISE – G. E. Maroney**
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SECTION VI.	**PUMP CONTAMINANT TOLERANCE VERIFICATION – L. E. Bensch**
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SECTION I

STRUCTURAL ANALYSIS OF CYLINDERS

PROJECT STAFF

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FOREWORD

This section presents a detailed account of the project activities in the area of cylinder dynamic strength evaluation. The relation of fatigue tests on material specimens to fatigue failure of hydraulic components such as cylinders is explicated, with particular reference to the two types of tests which are used for cylinder evaluation. Dynamic models for stress calculations are presented to show the effect of test system parameters. Experimental data are presented to show correlation between pressure and strain rise rates.

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CHAPTER I

INTRODUCTION

The reliability of hydraulic components can be appreciably enhanced by insuring that rational test procedures are used to evaluate the components before they are used for a specific application. From the outset of the association between the U.S. Army Mobility Equipment Research and Development Center and the Fluid Power Research Center of Oklahoma State University, the underlying objectives of all efforts undertaken have been directed towards developing and appraising test methods and procurement specifications so that they are consistent with the latest technological developments.

As part of this overall effort, the Fluid Power Research Center undertook to scrutinize current test procedures for structural integrity assessment of mobile hydraulic cylinders (i.e., non-tie rod types) under dynamic loading. Currently, there are two methods of performing such tests – first by stroking the rod back and forth and using a slave cylinder to impose the required load on the test cylinder and, second, by mechanically positioning the rod at a prescribed extension and alternately pressurizing opposite sides of the cylinder. These two methods are sometimes referred to as the *"stroking"* and *"impulse"* methods, respectively. Even though it is recognized that the stroking method is more realistic of actual cylinder applications, the impulse test is more attractive for product qualification and quality control, since for a given number of cycles it can generally be performed much faster. The question naturally arises as to whether the stroking test can be supplanted altogether. Another related question is whether the cycles to failure of the two tests are at all comparable. The answering of these questions requires first an appraisal for the failure criteria, second a dynamic analysis of the component stresses in the two test configurations, and third a study of the influence of test parameters on test results. It is appropriate to consider each of these aspects.

Hydraulic cylinders are heterogeneous components, since generally they use different materials for the cylinder body, end caps, piston rod, piston, seals, etc. Some of these materials are subject to gradual degradation which is reflected in cylinder performance (e.g., seal leakage is evidenced by cylinder drift or inability to hold loads). Other materials fail due to cyclic stressing in an essentially random manner. The failure of cylinder walls, rod eyes, pins, and most other metallic parts are examples of such failure, known in common engineering terminology as fatigue. It needs to be emphasized that because failure is random it does not imply that the part is as likely to fail after 10 cycles as after 10^5 cycles. It does imply, however, that probabilistic descriptions are needed to describe the failure (e.g., 99% confident that failure will not occur below 10^5 cycles). Fatigue failure in metals has been the subject of extensive studies, mostly empirical, for the last six decades. Chapter II presents a brief review of notable accomplishments and how they influence hydraulic component evaluation.

The second influential aspect noted earlier is that of dynamic stress analysis for the two test configurations – namely stroking and impulse tests. Since stresses are necessarily dependent on forces on various parts of the test cylinder, a dynamic mathematical model of each test configuration has to be developed as a prerequisite for any dynamic stress analysis. Fatigue failure is always related to local stresses and strains; but, in testing components, direct measurement of such stresses or strains is often impracticable. In hydraulic cylinders, the stresses are related to pressures and loads which are more easily measured. Chapter III shows how significant variables (pressures, flow rates, and principal stresses) change during both stroking and impulse tests.

Test system parameters can have significant influence on the pressure wave forms to which the test cylinders are subjected. The condition of the test cylinder (especially leakage and drag characteristics), the stiffness of the test frame, and the inertia of the moving parts are examples of test parameters which need to be controlled in order to have uniform testing. The influence of many of these parameters can be assessed before testing is commenced. Chapter IV shows how stress rise rates may be calculated from a knowledge of test system parameters, while Chapter V presents simulation results for a variety of test conditions.

When using the stroking and impulse tests for product qualification, it is desirable to keep the experimental procedure as simple as possible. Since pressure measurement is generally straightforward as compared to strain measurement, it is useful to establish the correlation between the two so that each test cylinder need not be instrumented for strain measurement. With this objective, a series of dynamic loading tests were performed to measure pressure and cylinder strain levels simultaneously. Chapter VI presents the results of these tests.

A review of pertinent paragraphs of the draft military specification for appraising mobile hydraulic cylinders is presented in Chapter VII. The paragraphs under review are included as Appendix A, while a list of test reports used for extracting historical test information on stroking and impulse tests is given in Appendix B.

CHAPTER II

AN OVERVIEW OF FATIGUE TESTING VIS-A-VIS HYDRAULIC COMPONENT EVALUATION

INTRODUCTION

The failure of metals by fatigue is an extremely common occurrence. In fact, a majority of material failure in engineering components or structures can be ascribed to fatigue, acting either alone or in conjunction with other causes (like corrosion, thermal stresses, etc.). Hence, the design of components to withstand dynamic loading requires a working knowledge of the mechanism of fatigue, properties of materials, and principles of fatigue-resistant design. Fatigue evaluation is still very much an empirical science (i.e., it is not possible to estimate with any degree of accuracy the fatigue life of a component like a hydraulic pump or cylinder merely by a scrutiny of the drawing, knowledge of material properties, and possibly knowledge of the operational duty cycle). Consequently, the conduction and interpretation of tests form an important part of fatigue evaluation.

Fatigue is an intrinsically stochastic phenomenon (i.e., there will always be a degree of variability in test data, even for the best of experiments). The interpretation of such data can only be done in probabilistic terminology, which though intuitively understood by many users can be the source of serious misunderstandings. In particular, the misuse of statistical formulae can lead to untenable conclusions and should be scrupulously guarded against.

Since fatigue analysis is still an empirical science, mathematical models used to describe fatigue phenomena are based almost entirely on experimental data. Expediency has often dictated the use of experimental setups used to obtain the data. Consequently, in using any

formula based on such data, every effort should be made to insure that it is valid for the intended application. Such a critical assessment is needed even of such *"universally"* accepted items as S-N diagrams, Goodman charts, and Miner's rule.

An important point to note is that these design aids are primarily meant to guide material selection. They cannot be expected to be used to predict the fatigue life of components for the following reasons:

1. **The S-N diagrams and similar charts are almost always developed for materials in the form of test specimens. It is difficult, if not impossible, to relate the nominal stress levels used for such tests to gross loads or pressures in components.**

2. **Components are generally heterogeneous and often built up of parts bolted or fastened together. Even if the connections are nominally rigid, there will be some relative motion and this can substantially change the stress cycles perceived by each element. Consequently, failure may occur in different elements, and the combined effect may bear little relation to coupon tests.**

3. **Stress concentration effects in components are very difficult to isolate and analyze. *Formulae for compensating* for such effects are empirical and rely heavily on standard tests using specified configurations.**

Before discussing component testing, it is useful to briefly review the coupon tests which have been and are being widely used to acquire fatigue data.

COUPON TESTING

Ever since Wohler performed his pioneering experiments about 100 years ago, engineers and scientists have been building and using fatigue testing machines. The rotary bending machines were the first to be widely used, since it was easy to subject specimens to a large number of stress reversals in a very short time. Wohler showed that (for the testing conditions he used) it is the number of stress cycles rather than the elapsed time of testing that is important

[Ref. 1]. Since then, it has been the unquestioning practice of researchers to report their results in the form of S-N (stress versus number of cycles) diagrams. Only recently, with the spotlight on low cycle fatigue and plastic deformation, have other methods of data presentation been even considered [Ref. 2]. Since the rotary bending fatigue tester is the most commonly used machine, it is instructive to review its mechanism so as to better understand the significance of test results. Fig. 2-1 shows a schematic arrangement of the beam type rotary bending fatigue tester.

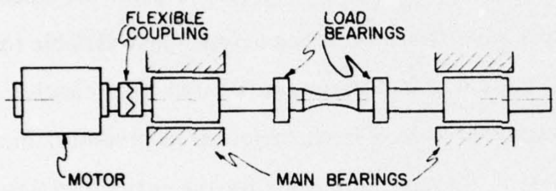

Fig. 2-1. Schematic Arrangement of Rotary Fatigue Testing Machine.

If the loading is symmetrical, the test section is subjected to pure bending; and, as the beam is rotated, portions of the test sections are alternately subjected to tensile and compressive stresses in one rotation. The test section is almost always round; consequently, the material is subjected to a sinusoidal stress cycle if the motor is driven at a uniform speed. Consequently, these machines (without elaborate modifications) can be used only for: (a) complete reverse bending, (b) symmetrical stress, and (c) periodic/cyclic loading (i.e., not stochastic).

It is also of interest to note that the stress distribution in the specimen is non-uniform, the material nearer to the axis being subjected to lesser stress than the surface material. Results from fatigue tests using the above machine should always be interpreted keeping the above points in mind.

Testing machines which use pure bending, axial stressing, torsion, and various combinations thereof have been evolved. The number of cycles for a given nominal stress exhibits significant differences between the machines [Ref. 1]. Since fatigue failure is one of crack generation, and growth and this in turn depends upon the stress distribution at the cross-section in question, such differences are not unexpected.

Specific mention must be made of low cycle fatigue testing machines. These were designed specifically to evaluate the fatigue strength for fairly short lives – Coffin defines it as 50,000 cycles [Ref. 2]. Such machines usually impose a predetermined *strain* cycle rather than a stress cycle. In view of the need for a feedback mechanism to control the loading, the cycling rate is usually 500 cycles/minute or less. Plastic strain is an extremely important factor in low cycle fatigue, and hysteresis loops of cyclic stress versus cyclic strain are often plotted to see if the material hardens, softens, or creeps.

A few remarks about the S-N curve are also in order. First, the quantity plotted on the ordinate is normally the peak alternating stress, even though it is the independent variable for the tests. Second, the stresses depicted are usually nominal stresses calculated from elastic strain theory. Third, the test specimen is considered to have been subjected to predetermined cyclic stress, or more strictly speaking loading, of the same amplitude for the entire duration of the test. (These remarks refer to conventional or high cycle testing only. Presentation of data on low cycle testing is more varied and usually much more complete.)

The main purpose of fatigue testing on standardized coupons is to obtain data which can be used as a guide for predicting the behavior of materials in service [Ref. 3]. Since, in many instances, the stress cycle on a component or a part of it may not even approximate the stress cycle used in generating the test data, it is obvious that life prediction based on the S-N curve alone is risky at best.

The rationale for fatigue testing on standardized specimens includes:

1. **Reduction in the number of uncontrollable variables and their influence, so that a better understanding of the basic mechanism may be obtained.**

2. **Examination of the influence of various factors (e.g., heat treatment, stress concentration, mean stress levels, etc.) in isolation and in combination.**

3. **Establishment of the statistical characteristics of fatigue data. (It. is much harder to obtain equally voluminous data on specific machines and components.)**

Perhaps the first use of S-N diagrams was to establish the fatigue limit – the stress below which a specimen of the material has an infinite life. Since it is impossible to test a specimen beyond, say, 10^{10} cycles, the fatigue limit is now defined as *"the limiting value of the median fatigue strength as N (number of cycles) becomes very large"* [Ref. 4]. Designers were then instructed to keep loads within fatigue limits to avoid fatigue failure. It was discovered that many materials, notably nonferrous alloys, do not display a well-defined fatigue limit, and the concept of designing for finite fatigue life is now replacing the earlier criterion. If actual loading were periodic and of the same peak magnitude for all cycles, the estimation of fatigue life for a given stress level would be a trivial operation using the S-N curve. Consider now the simplest deviation that can be met in practice – a shift of the mean stress level. The effect of a non-zero mean stress is to decrease or increase the allowable alternating stress, depending on its sign. Plots of alternating stress vs. mean stress or maximum vs. minimum stress, with cycle life as the parameter, are commonly used to present such data [Refs. 1 and 3]. The use of such diagrams is generally straightforward if stochastic considerations are not introduced. According to Juvinall [Ref. 1], as long as the maximum and minimum stresses are the same, the wave form and cycle frequency are of no consequence.

The second deviation from conventional testing conditions arises due to changes in stress amplitude over the life of a component. This is conventionally handled by applying Miner's *"Law"* of cumulative damage. According to this hypothesis, the application of n_1 cycles at a stress range S_1, for which the number of cycles to failure is N_1, causes a *"fatigue damage"* n_1/N_1, and failure will occur when

$$\sum \frac{n}{N} = 1$$

Though widely used, Miner's *"Law"* is not universally accepted, with at least one author [Ref. 5] contending *"There is, as it turns out, plenty of experimental evidence that this*

criterion is invalid, but it can serve as a rough approximation." How rough is illustrated by the value of $\sum n/N$ obtained in Miner's original tests – 0.7 to 2.2 [Ref. 1], while according to the law it should be 1. A serious drawback of the Miner cumulative damage rule is that it ignores the sequence in which cycles of different amplitudes are applied. Some materials for which stressing just below the endurance limit is followed by higher amplitude stress cycles have shown improvements in fatigue life [Ref. 3]. Miner's cumulative damage theory, with minor modifications, is still the most widely used basis for estimating life under varying stress conditions.

In order to make specimen testing more realistic, methods in which the stress level is changed in the course of a test have been introduced. One procedure, known as "*program testing,*" subjects the specimen or component or machine to a predetermined sequence of cycles, the amplitude of which may range from 0.1 to 1.0 times the maximum stress amplitude [Refs. 3 and 6]. Though such tests are more realistic than single stress level tests, evidence that they can simulate random loading is not conclusive [Ref. 3]. They also, however, show that Miner's "*Law*" often results in significant overestimates of life [Ibid].

The introduction of stochastic loading in fatigue testing, while it is intuitively appealing, raises questions of data interpretation which cannot always be objectively resolved. Stochastic loading can be mathematically described as random processes. However, only in a few applications (e.g., aircraft gust loading) have enough test data been collected to describe the random process in probabilistic terminology. Even here, it is generally assumed that it can be described by its power spectrum and auto-correlation function. This information is then used to generate a test specimen random loading, which has the same statistical properties as the measured actual loads. Since the generation of truly random loading to conform with specified statistical properties (typically stationarity, a given power spectrum, and a given auto-correlation function) is not easily done, some researchers have used "*pseudo-random*" loading, typically sinusoidal loading (which is modulated by a deterministic or random signal [Refs. 6 and 8]. As can be anticipated, the interpretation of such test data is not always noncontroversial.

Mention must be made of methods of using random load data with cumulative damage assessment procedures. The problem here is of demarcating individual cycles. At least four different methods have been proposed [Ref. 9], and they do not always yield the same results. As long as the sequence of different cycles is ignored, reconciliation between the different counting techniques will be only of limited use for life estimation.

COMPONENT AND MACHINE TESTING

Even though specimen testing on conventional rotating bending, axial, or torsion machines can give the machine designer a good idea of the fatigue resistance characteristics of a material and the influence of certain general factors (such as surface finish, heat treatment, key slots, etc.), such testing by itself cannot be used for the life estimation of critical components and design evaluation where fatigue considerations are of primary importance. In aircraft and automotive applications, it is common practice to subject components and entire assemblies to fatigue tests [Refs. 3 and 6]. Attempts are made to simulate actual working conditions, but the test results can rarely be used for drawing general conclusions. They do, however, spotlight the weak spots in a design and prove its worthiness to licensing/regulatory agencies and equipment buyers.

The design of tests for assessing the fatigue strength of components and machines is beset by many difficulties. First, components and machines are heterogeneous, and sufficient data from different failure modes are hard to collect. Second, it is often hard to distinguish the failure mode and the critical element in an assembly, especially when they interact with each other. Third, the design of accelerated tests is difficult if not impossible. Cyclic loading rates cannot be altered as easily as in coupon testing. Physical limitations place constraints on pressure and stress rise rates, wave shape, etc. Fourth, statistical analysis is much more complex and harder to interpret, in view of the generally scanty test data coupled with the number of uncontrollable disturbing factors. It is no surprise then that a number of component and machine fatigue evaluation specifications contain almost arbitrary requirements.

HYDRAULIC COMPONENTS

The use of fatigue tests on standard specimens should be no different in the fluid power industry than in other industries using the same materials. As mentioned earlier, such tests, though useful in selecting materials, evaluating the general influencing factors like notch sensitivity, heat treatment, grain size, etc. cannot be used for predicting the fatigue life of components like pumps, valves, cylinders, motors, etc. Some of the factors that are responsible for this are:

1. **Complex geometries, making it difficult to calculate point stress.**
2. **Complex stress, quite often tri-axial.**
3. **Complex failure mechanism with much interaction between different parts of the component.**

Consequently, fatigue tests on components and assemblies and even complete machines must be seriously considered if fatigue failure is considered significant. Here, the potential for damage by failure and its costs have to be weighed against the cost of conducting the tests. In some instances (e.g., the aircraft industry), such tests are mandatory. Each new design and each new machine has to be tested anew; and, though previous testing experience is useful, it cannot supplant systematic testing. In such tests, a compromise has to be made between the desire for *"realistic"* testing and quick testing. The latter is usually resorted to by using the statistical description of actual loading conditions to generate pseudo-random loads, which can be used on testing machines. This procedure should be seriously considered for hydraulic components subject to random loading. It is particularly important to note that each such specialized test results in a population of raw data for statistical analysis, and such results from entirely different components, testing machines, or test materials should not be merged without appraising the implications in statistical analysis.

Though mechanical forces can give rise to cyclic stresses in hydraulic components, the more common stressing mechanism is pressure. Almost all hydraulic components serve as pressure vessels (in addition to other functions), hence the importance of evaluating their

fatigue resistance to pressure pulsations. A common feature of pressure vessels is that, in the absence of pre-stressing, they are generally subjected only to tensile stresses. Also, the stressing is usually tri-axial and complex.

In hydraulic cylinders, the cylinder wall is always subjected to hoop stresses which are proportional to the cylinder pressure. Axial stresses should be present only when the cylinder is completely extended or retracted, since cylinder drag forces are normally small compared to external loads. Bending stresses generally arise due to self-weight and external moments. Such moments are often introduced as a result of misalignment of pins. Cylinder rods are generally subjected to axial stresses (compressive or tensile, depending on loading) and bending stresses for the same reasons as given above.

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CHAPTER III

HYDRAULIC CYLINDER FATIGUE LIFE EVALUATION

INTRODUCTION

There are two tests which are being used currently to evaluate the fatigue strength of hydraulic cylinders. In the first, which will be referred to as the *"locked rod"* or *"impulse"* test, the rod is positioned at midstroke and rigidly held there while pressure is applied alternately on either side of the piston. In the second, which will be called the *"cycling"* or *"endurance"* test, the rod traverses back and forth, a specified load being maintained at all times. It is desired to develop a rational basis for comparing the results from these two tests so that testing can be accelerated.

IMPULSE TEST

Fig. 3-1 presents a circuit schematic of the prescribed test setup. It is seen that the cylinder is rigidly fixed at two points, and the only relative motion of the rod and cylinder is due to the flexure of the structure.

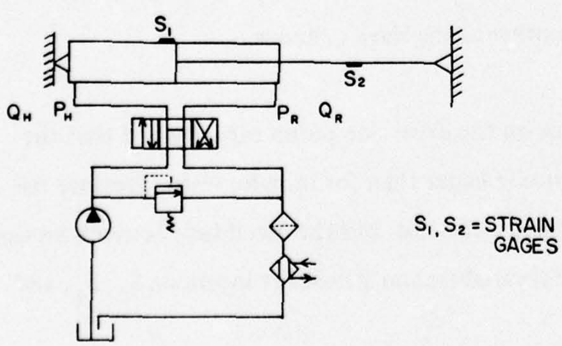


Fig. 3-1. Circuit Schematic for Impulse Testing.

It is seen that the cycling rate depends on the switching speed of the directional control valve. The pressure rise and fall rates depend on the pump capacity, the elasticity of connecting lines, the bulk modulus of the fluid, and the switching time

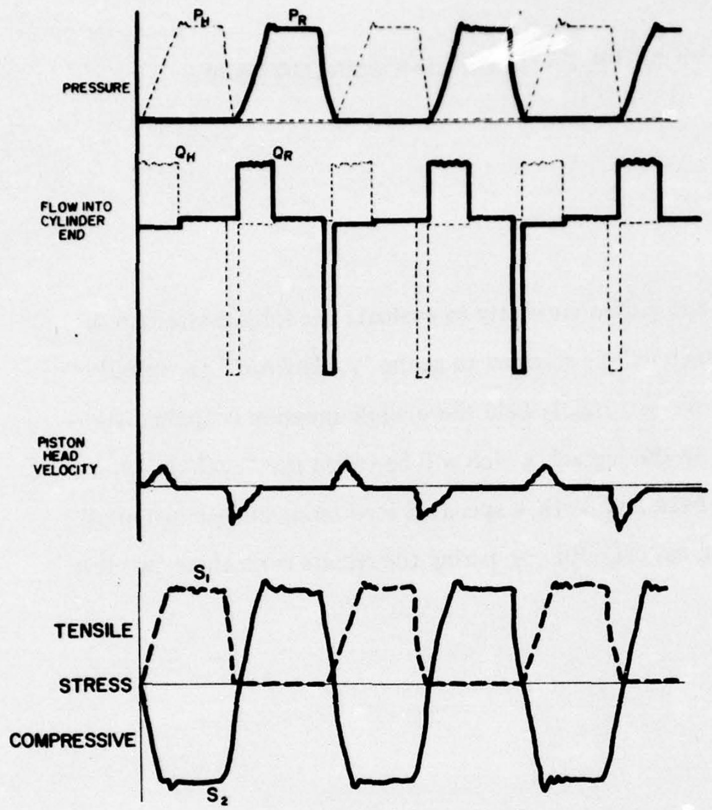


Fig. 3-2. Theoretical Traces of Variables in Impulse Testing.

of the directional control valve. Fig. 3-2 depicts the idealized traces of significant physical variables and the accompanying cylinder hoop stress at S_1 and rod stress at S_2 (in Fig. 3-1).

ENDURANCE TEST

Fig. 3-3 presents a circuit schematic of the test setup. It is the normal practice to test two identical cylinders back to back, using one as the driving cylinder and the other as the slave cylinder.

It is seen that the cycling rate is dependent on the drive side pump capacity and that the cycling time will, in general, be orders of magnitude larger than for impulse tests. Pressure rise and fall rates may be comparable to those in the impulse test, but the dwell time is much longer. Fig. 3-4 presents the idealized traces for physical variables and stresses at locations S_1, S_2, and S_3.

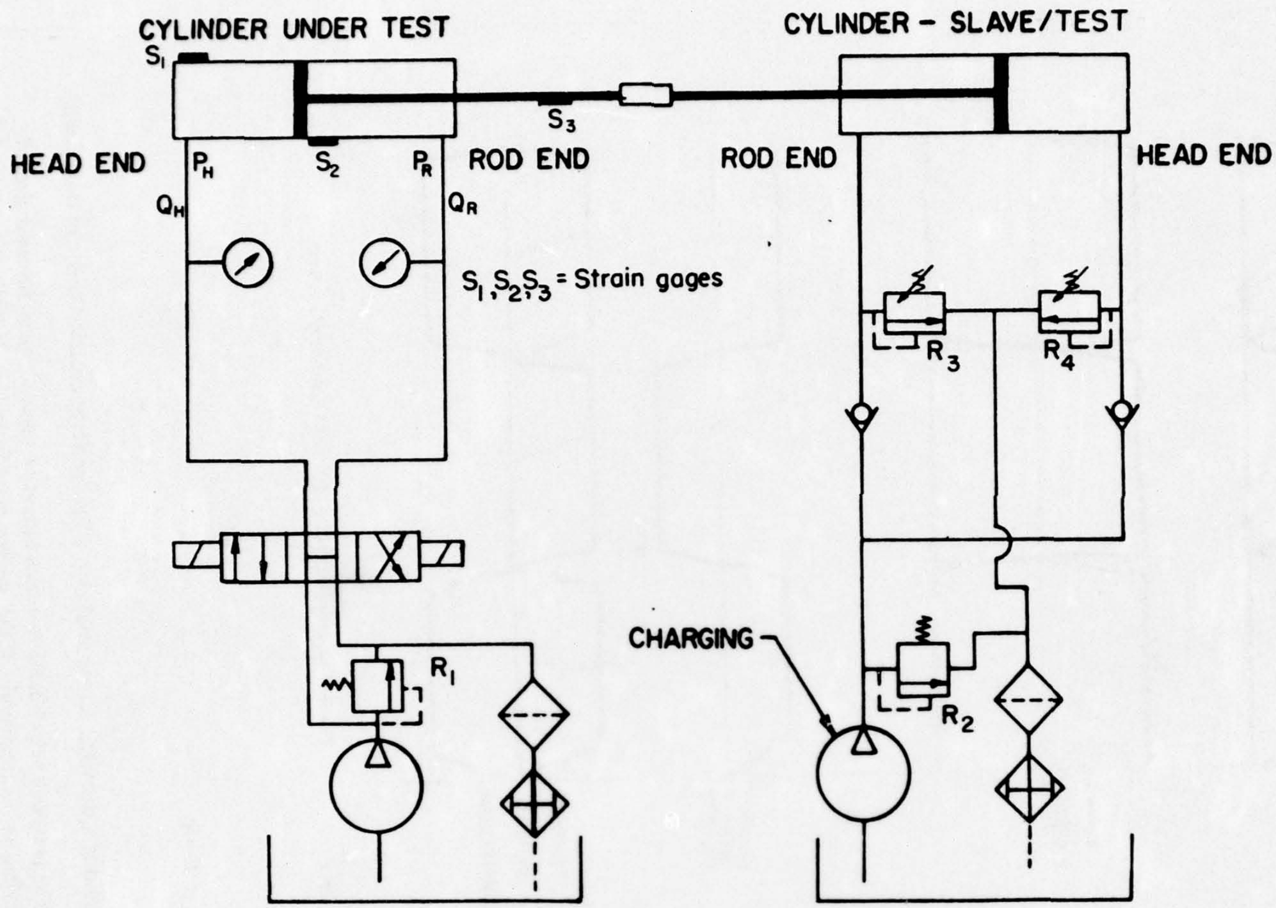


Fig. 3-3. Circuit Schematic for Stroking Test.

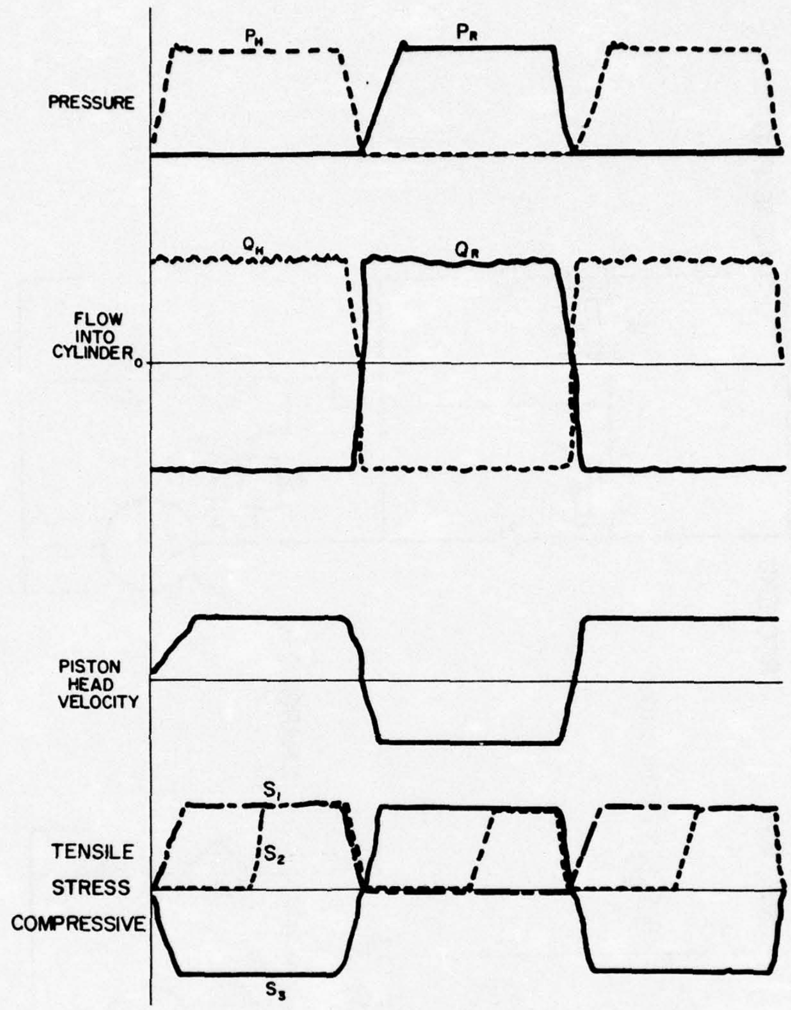


Fig. 3-4. Theoretical Traces of Variables in a Stroking Test.

Comparison of Tests

Even without a detailed stress analysis, a few general conclusions can be drawn:

1. **The pressure cycles for the two tests cannot be made identical for most sizes of cylinders. Consequently, if the pressure wave form and especially the time under pressure are different for each test, the number of cycles to failure can be expected to differ.**

2. **The impulse test is unrealistic as far as wear of rod and seals is concerned. It is, however, a realistic test to assess the fatigue strength of the rod.**

3. **The cylinder is not subjected to similar stressing in the two tests. In the impulse tests, the cylinder walls on either side of the piston are alternately subjected to hoop stress; whereas, in the endurance test, in one cycle, a point on the cylinder wall experiences the maximum stress for different times, depending upon its position. A comparison of stress traces shown in Figs. 3-2 and 3-4 makes this clear.**

A more detailed analysis, based on strain energy considerations needs to be taken to ascertain the precise relations between the two tests. A conclusion that can certainly be drawn at this time based on the above points is that the two tests are not comparable.

Test Data Evaluation

Some test data on impulse and endurance tests performed as described above are available at this time. These tests were performed not so much to compare the two tests as to qualify the cylinder designs for certain types of equipment. Table 3-1 furnishes a summary of the test results. An inspection of the data reveals the following:

1. **Multiple samples of similar cylinders were not tested.**

2. **If a prescribed number of cycles were completed, the tests were terminated (i.e., testing was not carried out to failure). Hence, if a cylinder qualified in both tests, no conclusion as to comparative severity can be drawn.**

3. **Failure modes are widely different. In some tests, cylinders were repaired and even modified after initial failure and the test continued. In fact, the decision as to whether a given cylinder qualified seems to have been subjective in at least one or two cases.**

It should be emphasized that the above points do not constitute a criticism of the individual qualification. They do, however, make it difficult to compare the two tests from a severity standpoint.

TABLE 3-1. SUMMARY OF TEST RESULTS ON CYLINDERS.

Test No.	Cylinder Size (in.)	Pressure (psi)	Locked Rod		Stroking Rod		Comments
			Cycles	Failure	Cycles	Failure	
(1)	4	2000	750,420	Nil	750,000	Nil	
(2)	6	2000	254,908	Piston Rod	296,844	Piston Rod	
(3)	8	2000	113,850	Piston Rod	203,108	Eyebolt	
(4)	4.5	1975	750,000	Nil	750,000	Nil	
(5)	3.5	2000			20,673	Seal Leakage	
					650,100	Nil	(after repairing seal)
(6)	4.5	2000			272,720	Leakage	No Failure.
(7)	3.5	2000			1,032,327	Piston Rod	
(8)	3.5	2000			64,654	Seal Leakage	
(9)	4.5	2000			22,846	Seal Leakage	
					488,114	No Failure	
(10)	5.0	2000			195,950	No Failure	
(11)	3.75	2000			543,345	No Failure	
(12)	5.0	2000	157,209	Pin Failure	151,913	Pin Failure	
			239,759	Yoke Elong.			
			302,874	Pin Failure			
(13)	4	2000	750,077	Nil	750,000	Nil	
(14)	8	2000	86,400	Piston	24,564	Piston	Taper Failure
(15)	6.5	2000	46,190	Rod Failure	71,218	Rod Yoke	Repaired
			192,472	Rod Failure	76,141	Seal Failure	Repaired
					105,007	Rod Yoke	
(16)	8.5	2000	22,058	Cap Bolts	106,400	Cap Bolt	Repaired
			145,503	Rod Failure	146,962	Cap Bolt	Repaired
					248,747	Cap Bolt	Repaired
					255,756	Rod Eye	Repaired
					457,389	Stop Nut	Repaired
					468,686	Seal Failure	Repaired
					507,805	Stop Nut	Repaired
					573,918	Seal Failure	Repaired
					587,262	Rod Failure	Repaired
(17)	4.5	2000	191,345	Rod Eye	750,000	Nil	
			362,273	Bushing			Repaired
			800,000	Nil			
(18)	4	1500	347,316	Rod Failure			
			95,907	Flange Break			Repaired
			750,000	Plating			
			123,978	Rod Failure			Repaired
			300,000				Flange Changed
			750,000	No Failure			
(19)	5	2000	420,682	Cylinder (after stroking)	58,879	Bolt Failure	Repaired
					1,000,000	No Failure	
			1,000,000	No Failure			(after stroking test)
(20)	4	2000	98,286	Rod Failure (after stroking)	750,000	No Failure	
(21)	4	2000	580,941	Rod Failure (after stroking)	750,000	No Failure	
(22)	5	2500	7,000	No Failure	3,000	No Failure	

CONCLUSION

The impulse and endurance tests on cylinders are not comparable for *all* parts of a cylinder. A restricted comparison of severity can be done after an analysis of the strain histories in the rod and cylinder wall. Most of the failures seem to occur at the pins or yokes, pointing out the importance of assembly and alignment. Tests not run to failure are of little use for

comparison purposes. Tests in which design changes are made after initial failure are also of little value beyond that point. Further analysis based on strain energy considerations is needed to indicate the severity of each test.

CHAPTER IV

A DYNAMIC MODEL FOR STRESS CALCULATIONS DURING STROKING TESTS

NOMENCLATURE

SYMBOL	DESCRIPTION
A	Area
B	Drag Coefficient
I	Inertia Coefficient
k	Relief Valve Constant
m	Relief Valve Gradient
P	Pressure
V	Volume
x	Cylinder Movement
β	Effective Bulk Modulus of Fluid

SUBSCRIPTS	
A	Head End
B	Rod End
S	Slave Cylinder
T	Test Cylinder

The purpose of this mathematical model is primarily to ascertain the pressure and stress rise rates which can be obtained with a prescribed test setup. The endurance test stand shown schematically in Fig. 3-3 is the basis for this analysis. Pressure rise rates are sometimes specified in conjunction with flow rates for the test cylinder circuit pump, and these two could be in conflict under certain circumstances.

The following constraints, which can be relaxed when a more accurate treatment is desired, are made for a preliminary analysis:

1. **The four-way valve in the test cylinder circuit offers no flow resistance.**
2. **The test circuit pump delivers the same flow rate at all pressures.**
3. **The guide in which the two cylinder rods are connected is frictionless.**
4. **The relief valves on the slave cylinder circuit have a straight line pressure-flow characteristic. They do not leak below the cracking pressure and do not exhibit hysteresis.**
5. **There is no leakage past the pistons in either the test or slave cylinder.**
6. **Lines offer no resistance to flow.**
7. **Back pressure in both test and slave cylinders is zero.**

The equations governing the system variables are as follows:

$$Q_T = A_{AT}\dot{x} + \dot{P}_{AT}\frac{V_{AT}}{\beta} \qquad (4\text{-}1)$$

$$P_{AT}A_{AT} = I\ddot{x} + (B_T + B_S)\dot{x} + P_{AS}A_{AS} \qquad (4\text{-}2)$$

$$A_{AS}\dot{x} = \dot{P}_{AS}\frac{V_{AS}}{\beta} + (k_S + m_S P_{AS}) \qquad (4\text{-}3)$$

Eq. (4-1) is obtained from compressibility and flow considerations of the test cylinder circuit. Eq. (4-2) is obtained by performing a force balance on the moving parts. Eq. (4-3) is obtained from compressibility and flow considerations of the slave cylinder circuit. It should be noted that Eq. (4-3) is valid in its present form only if P_{AS} is greater than the cracking pressure of the relief valve.

Though it is possible to numerically solve Eqs. (4-1), (4-2), and (4-3) on the digital computer, better insight into the dynamic phenomena is obtained by first studying them qualitatively. Such a study follows.

At the beginning of a stroke, the relief valve in the slave cylinder circuit will be shut; consequently, Eq. (4-3) reduces to:

$$\dot{P}_{AS} \frac{V_{AS}}{\beta} = A_{AS} \dot{x} \tag{4-4}$$

This can be integrated to give:

$$P_{AS} \frac{V_{AS}}{\beta} = A_{AS} x \tag{4-5}$$

since we can assume that $P_{AS} = 0$ when $x = 0$.

Substituting the value of P_{AS} given by (4-5) in Eq. (4-2), we get:

$$P_{AT} A_{AT} = I\ddot{x} + (B_T + B_S)\dot{x} + \frac{A_{AS}^2 \beta}{V_{AS}} x \tag{4-6}$$

Differentiating Eq. (4-6) with respect to time, we get:

$$\dot{P}_{AT} A_{AT} = I\dddot{x} + (B_T + B_S)\ddot{x} + \frac{A_{AS}^2 \beta}{V_{AS}} \dot{x} \tag{4-7}$$

Substituting for $\dot{x}$, $\ddot{x}$, and $\dddot{x}$ by using Eq. (4-1) [differentiating it for $\ddot{x}$ and $\dddot{x}$], we get:

$$\begin{aligned} \dot{P}_{AT} A_{AT} = {} & I \left[- \frac{\dddot{P}_{AT} V_{AT}}{\beta A_{AT}} \right] + (B_T + B_S) \left[- \frac{\ddot{P}_{AT} V_{AT}}{\beta A_{AT}} \right] \\ & + \frac{A_{AS}^2 \beta}{V_{AS}} \left(Q_T - \frac{\dot{P}_{AT} V_{AT}}{\beta} \right) \frac{1}{A_{AT}} \end{aligned} \tag{4-8}$$

or

$$\begin{aligned} \left[A_{AT} + \frac{A_{AS}^2 V_{AT}}{V_{AS} A_{AT}} \right] \dot{P}_{AT} = {} & - I \frac{\dddot{P}_{AT} V_{AT}}{\beta A_{AT}} - (B_T + B_S) \frac{\dot{P}_{AT} V_{AT}}{\beta A_{AT}} \\ & + \frac{A_{AS}^2 \beta}{V_{AS}} \frac{Q_T}{A_{AT}} \end{aligned}$$

Since initially both $\dddot{P}_{AT}$ and $\ddot{P}_{AT}$ will be positive, it is seen that the pressure rise rate $\dot{P}_{AT}$ is lessened due to the presence of the inertia and drag terms. As an extreme situation, consider the case when $I = B_S = B_T = 0$.

$$\dot{P}_{AT} = \frac{A_{AS}^2 \beta Q_T}{V_{AS} A_{AT}} \left[\frac{V_{AS} A_{AT}}{V_{AS}A_{AT} + V_{AT}A_{AS}^2} \right] \tag{4-9}$$

As a numerical example, consider the following:

Test and Slave Cylinders:	**5" diameter by 24" stroke (127mm diameter by 0.6 m stroke)**
Dead Volume at End of Stroke:	**10 cu. in. (164 ml)**
Pump Flow Rate:	**20 gpm (75.7 lpm)**
Fluid Bulk Modulus:	**2 x 10^5 psi (13,794 bars)**

When the test cylinder is at the beginning of a stroke, the volumes in Eq. (4-9) are as follows:

$$V_{AT} = 10 \text{ cu. in.} \quad (164 \text{ ml})$$

$$V_{AS} = 471 \text{ cu. in.} \quad (7.718 \text{ litres})$$

$$\therefore \dot{P}_{AT} = \frac{(19.63)^2(2 \times 10^5)(20.385)}{(471 + 10)(19.63)^2}$$

$$= 32{,}016 \text{ psi/sec } (2200 \text{ bar/s})$$

In practice, the pressure rise rate will be less than this, primarily due to leakage past seals. Only a simulation of Eq. (4-8) will yield the time-history of the pressure rise rate as the cylinder is stroked. It is, however, of interest to estimate the pressure rise rate obtained by using the same flow rate on a locked rod test.

If the rod is locked in mid-position, the trapped volume is approximately (471/2 + 10) = 245.5 cu. in. (4.02 litres). Pressure rise rate is given by:

$$\mathring{P}_{AT} = \frac{Q_T \beta}{V_{AT}} = \frac{(20.385)\,(2 \times 10^5)}{245.5}$$

$$= 62{,}729 \text{ psi/sec} \quad (4{,}326 \text{ bar/s})$$

This demonstrates that, for a given flow rate, the locked rod test will generally result in higher pressure rise rates. The strain and stress rates in the cylinder and rod will be approximately proportional to the pressure rate. The maximum hoop stress in the pressurized portion of a cylinder is given by:

$$S_{\text{hoop max.}} \triangleq P_{AT} \left(\frac{R^2 + r^2}{R^2 - r^2} \right)$$

where: $R \triangleq$ External Radius of Cylinder

$r \triangleq$ Internal Radius of Cylinder

If in the above example the test cylinder had a wall thickness of 0.625" (15.88 mm), the maximum hoop stress rate would be:

$$\dot{S}_{\text{hoop max.}} = 62{,}729 \left[\frac{3.125^2 + 2.5^2}{3.125^2 - 2.5^2} \right]$$

$$= 286 \times 10^3 \text{ psi/sec} \quad (19.7 \times 10^3 \text{ bar/s})$$

It is normally impracticable to measure this stress, since it occurs at an internal surface. The external hoop stress, which is easier to measure, is given by:

$$S_{\text{hoop ext}} \triangleq p \frac{2\,r^2}{(R^2 - r^2)}$$

and the external hoop stress rise rate in the above example would be:

$$\overset{\circ}{S}_{hoop\ ext} = 62{,}729 \left[\frac{22.5^2}{3.125^2 - 2.5^2} \right]$$

$$= 223 \times 10^3 \text{ psi/sec } (15.4 \text{ bar/s})$$

The rod will be subjected to a compressive stress proportional to the cylinder pressure. It should be noted, however, that if there is any slack in the pins at either end both the pins and the rod will be subjected to impact loading and the transient contact stresses may be much higher than that indicated by the above estimate.

CHAPTER V

SIMULATION OF STROKING AND LOCKED ROD TESTS

Chapter IV presented estimates of the pressure rise rates for both stroking and locked rod tests. It was shown that, for a given flow rate, if leakage and test frame flexure were ignored, the locked rod test would generally result in higher pressure rise rates.

Computer programs have been written to completely simulate both stroking and locked rod tests. These are discussed in the next two sections. The simulation results will be found very useful, both in designing and conducting tests as well as interpreting test results.

STROKING TEST SIMULATION

Fig. 5-1 shows the essentials of the test fixture based on the circuit shown in Fig. 3-3.

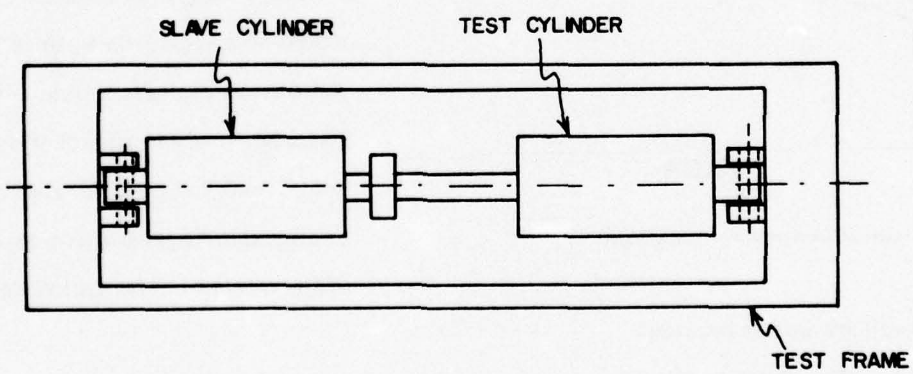


Fig. 5-1. Stroking Test Frame Schematic.

Usually, the test and slave cylinders are identical and tests are conducted on pairs of cylinders

which alternate as *"test"* or *"slave"* cylinders after a specified number of cycles. The cylinder traverse length is adjusted by limit switches, whose position may be changed according to a specified schedule. Since the test cylinder circuit pump has to deliver fluid alternately to the head and rod end, the *"time at pressure"* depends on the pump capacity in addition to the distance traversed per cycle.

The following assumptions were made in developing the mathematical model for the stroking test:

1. **Pump outflow is constant.**
2. **When one side of the test cylinder is pressurized, the other is at atmopheric pressure.**
3. **The directional valve switches instantaneously.**
4. **The relief valves in the slave cylinder circuit have characteristics, as shown in Fig. 5-2.**
5. **Cylinder drag force is proportional to velocity.**
6. **Cylinder leakage is proportional to pressure.**

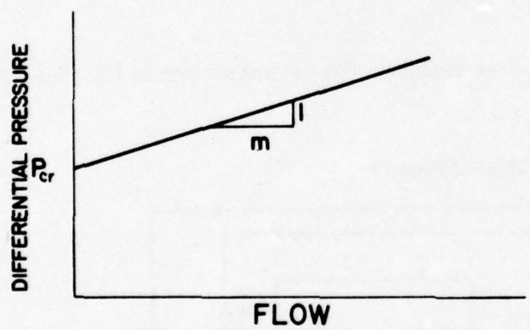


Fig. 5-2. Static Characteristics of Relief Valve.

It is useful to examine the behavior of the test system qualitatively before discussing the simulation results. Ideally, the pressure cycle would be a square wave, since the pressure on one side should rise instantaneously to the relief setting, hold during the traverse, and drop to atmospheric pressure on reversal of the directional control valve. The rise and fall are slower because:

(a) **The moving parts have inertia and need to be accelerated and decelerated at the beginning and end of traverse. Increasing the inertia tends to reduce the pressure rise rate, since it takes longer to accelerate to the final traverse velocity.**

(b) **The piston rods exhibit drag, which increases the test cylinder pressure to a level higher than that of the relief valves on the load circuit. Increasing this drag tends to increase the pressure rise rate in addition to raising the test cylinder final pressure.**

(c) **Both test and slave cylinders will generally exhibit leakage which normally increases as the seals wear out. Increase of slave cylinder leakage may result in reduction of the effective load on the test cylinder, and increase of test cylinder leakage will tend to reduce the pressure rise rate.**

Table 5-1 presents the test system parameter values used in digital simulation. The first of the three trends indicated above is evident in the simulation results summarized in Table 5-2 and depicted in Fig. 5-3(a)-(d). (Computer run 576 shows lower rise rate than 568, and the latter is higher than 562.)

TABLE 5-1. TEST SYSTEM PARAMETERS FOR STROKING TEST SIMULATION.

Name			Value	
Computer	Algebraic	Description	US Units	SI Units
AAT	A_{AT}	Head End Area, Test Cylinder	$19.63\ in^2$	$126.6\ cm^2$
AAS	A_{AS}	Head End Area, Slave Cylinder	$19.63\ in^2$	$126.6\ cm^2$
DVOL1	V_{d1}	Dead Volume, Slave Cylinder	$10.0\ in^3$	164 ml
DVOL2	V_{d2}	Dead Volume, Slave Cylinder	$10.0\ in^3$	164 ml
BETA	β	*Effective Fluid Bulk Modulus*	2×10^5 psi	13.8×10^3 bars
VAT	V_{AT}	Pressurized Volume, Test Cylinder Circuit		
VAS	V_{AS}	Pressurized Volume, Slave Cylinder Circuit		
QT	Q	Pump Inflow, Test Cylinder Circuit	$173.25\ in^3/sec$	170.3 ℓ/min
RELFCR	P_r	Relief Valve Cracking Pressure, Slave	2800.00 psi	193 bars
XMS	m	Relief Valve Gradient	$0.866\ in^3/$ sec/psi	12.3 ℓ/min bar

Figs. 5-4(a) and (b) show the effect of increased leakage. These data are of particular interest, since one criterion for cylinder qualification is that, after cycling, the test cylinder should not exhibit excessive leakage. If the pressure cycles are recorded at frequent intervals during cycling, any sudden decrease in the rise rate could be used as an indication of excessive leakage, if other test parameters do not change.

TABLE 5-2. EFFECT OF PISTON ROD INERTIA AND SEAL DRAG ON PRESSURE RISE RATE IN STROKING TEST.

Computer Run No.	Piston Rod Inertia		Drag Coefficient*		Average Pressure Rise Rate	
	lbsf sec^2	kg	lbsf sec/in	NS/mm	psi/sec	bar/s
562	0.2	35	20	3.5	36,664	2529
568	0.2	35	50	8.76	44,070	3040
576	0.5	86.3	50	8.76	38,513	2656
589	0.5	86.3	20	3.5	30,373	2095
584	0.5	86.3	5	0.88	34,823	2402

*Same value used for test and slave cylinders.

In actual practice, the inertia of the piston is invariant and consequently changes in pressure rise rates can be attributed to the effectiveness of the seals. In order to put the seal drag and leakage figures in proper perspective, it may be noted that, for the simulated conditions:

1. A drag coefficient of 95.32 lbsf sec/in amounts to a pressure differential of 4.85 psi across a 5" cylinder at a velocity of 1"/sec.

2. A leakage coefficient of 1.417×10^{-7} in^3/sec lbsf corresponds to a cylinder drift of 1" per hour at 2000 psi, while a value of 6.407×10^{-8} in^3/sec corresponds to a cylinder drift of 0.46" per hour at 2000 psi.

NOTE: A ten-fold increase in the leakage coefficients for both test and slave cylinders has resulted in approximately 4% drop in the pressure rise rate.

LOCKED ROD (IMPULSE TEST) SIMULATION

Fig. 3-1 shows the test circuit schematic, and Fig. 5-5 depicts the test frame in which the cylinder is mounted. Note that the rod is locked in one position (usually at mid-stroke), so that the only relative movement between the cylinder and rod is due to slack in the pinned ends and the flexure of the frame itself. When the solenoid valve is switched from one position to another, alternative sides of the cylinder are pressurized. The relief valve insures that the

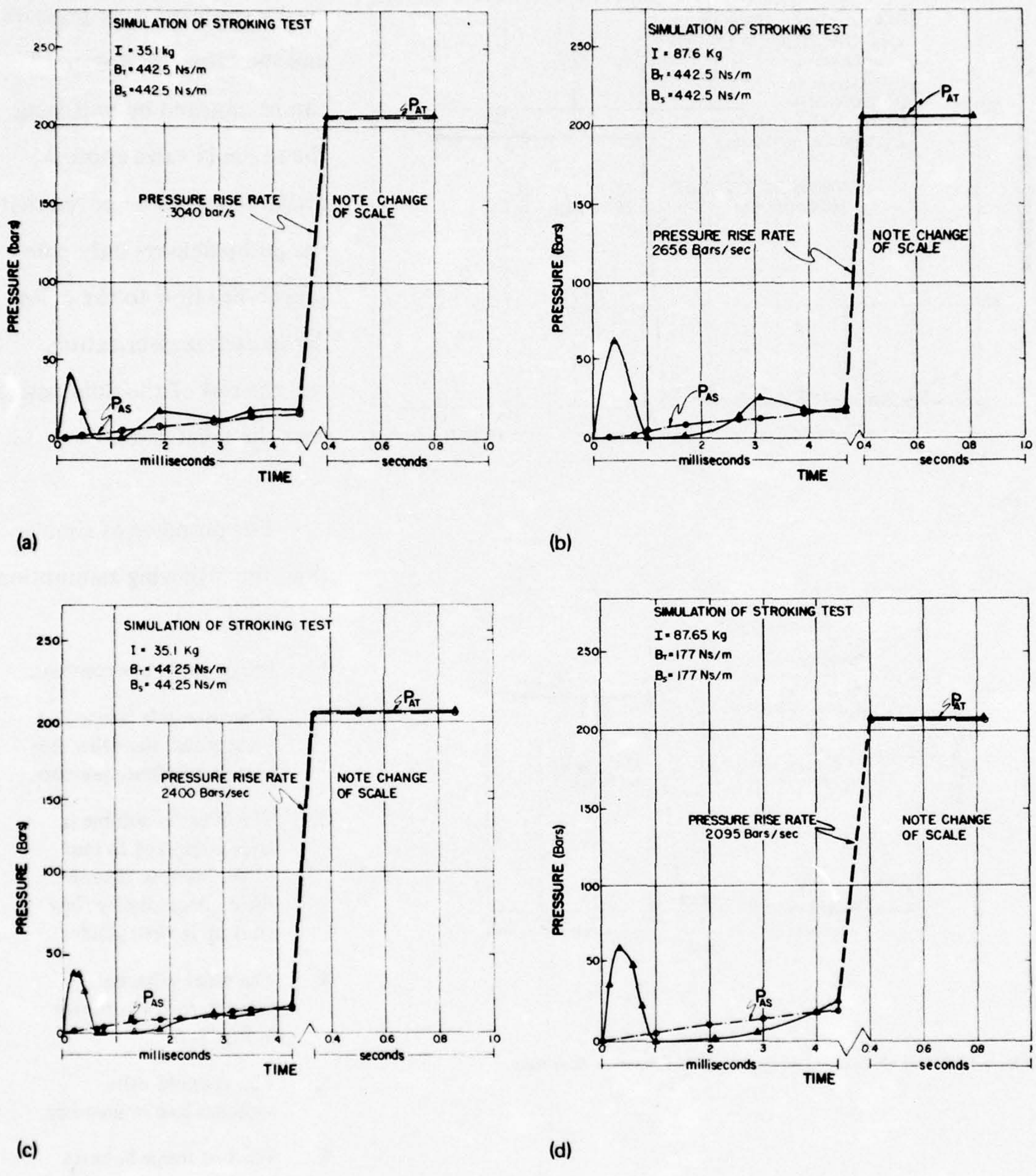
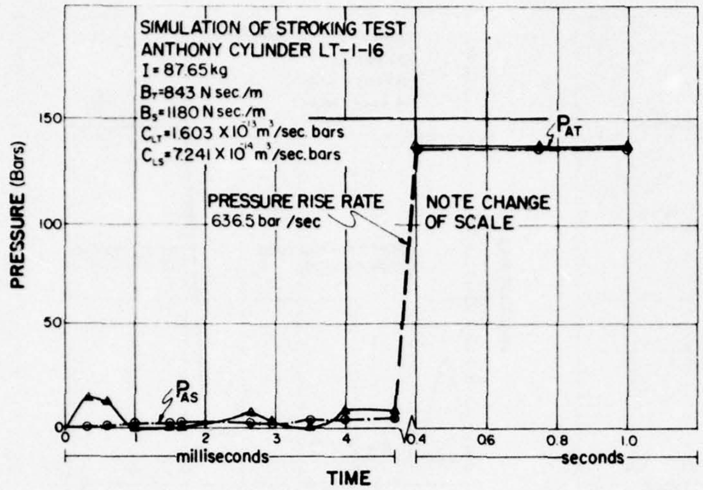
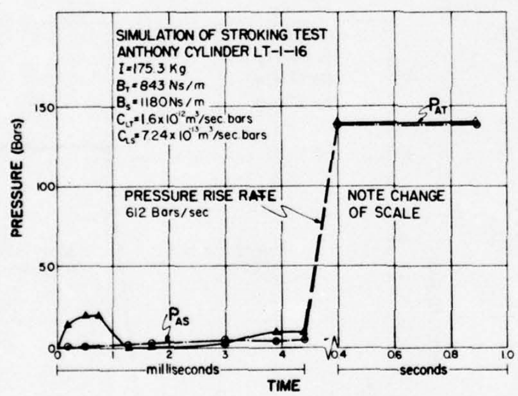


Fig. 5-3. Simulation Results for Stroking Test with Zero Leakage.



(a)



(b)

Fig. 5-4. Effect of Leakage on Stroking Test Pressure Rise Rate.

system pressure does not exceed a specified peak pressure, and the *"time on pressure"* can be adjusted by switching the solenoid valve appropriately. It should be noted that the pump delivers only compressibility flow to the cylinder in each pressurization, and the rest of the fluid passes over the relief valve.

For purposes of simulation, the following assumptions were made:

1. **Pump outflow is constant.**
2. **When one side is being pressurized, the other side is at atmospheric pressure.**
3. **The cylinder volume is large compared to that of the lines, so that all the compressibility flow ends up in the cylinder.**
4. **The relief valve has characteristics as shown in Fig. 5-2.**
5. **The solenoid valve switches instantaneously.**
6. **The test frame behaves like a linear spring.**

Special mention must be made of the following factors which are considered in simulation:

1. Inertia of the piston and equivalent inertia of the test frame are consolidated in one parameter.

2. Slack in the test frame, mainly due to clearances at the pinned connections. For purposes of simulation, the test frame is assumed to behave like a nonlinear spring, while this slack is being taken up. Once slack is taken up, the frame behaves like a linear spring.

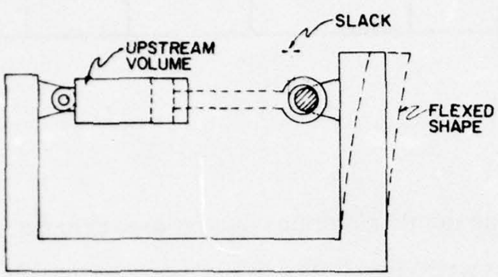


Fig. 5-5. Cylinder Impulse Test Frame Schematic.

Table 5-3 presents the numerical values of the parameters used in the simulation. The cylinder area corresponds to one of 5" (127mm) bore, while the upstream volume corresponds to approximately a 36" (914mm) long cylinder locked at mid-stroke. The volume of the piping between the pump and the cylinder is considered negligible by comparison. The drag coefficient, though included in the mathematical model, has an extremely small influence, since there is very little relative motion between the cylinder and the rod. The slack in the test frame results in low pressure rise rates in the beginning of a cycle, but once it is taken up the rise rate is controlled primarily by the frame stiffness and the pump flow rate.

TABLE 5-3. TEST SYSTEM PARAMETERS FOR IMPULSE TEST SIMULATION.

Name		Description	Value	
Computer	Algebraic		US Units	SI Units
AA	A_A	Cylinder Area	19.63 in^2	126.6 cm^2
VA	V_A	Upstream Volume	363.00 in^3	5.94 ℓ
QP	Q_P	Pump Flow Rate	40.425 in^3/sec	40 ℓ/min
BETA	β	Fluid Effective Bulk Modulus	2.0 x 10^5 psi	13.8 x 10^3 bars
RELFCR	P_{cr}	Cracking Pressure of System Relief Valve	1900.00 psi	131 bars
XM	M	Relief Valve Gradient (See Fig. 2-3.)	0.40425 in^3/sec/psi	5.8ℓ/min/bar
DRAG	B	Cylinder Drag Coefficient	133.44 lbsf sec/in	23.4 NS/m
CLRNCE	c	Slack in Test Frame (See Fig. 2-7.)	.008 in.	.203 min.

TABLE 5-4. EFFECT OF CYLINDER CONDITION ON PRESSURE RISE RATE IN IMPULSE TEST.

Computer Run No.	Frame Stiffness		Piston Rod Inertia		Cylinder Leakage Coefficient		Average Pressure Rise Rate*	
	lbsf/in	N/mm	$lbsfsec^2/in$	kg	$(in^3/sec/psi)$	ml/sec bar	(psi/sec)	bar/s
282	2.93×10^5	69×10^3	0.5	87.6	1.417×10^{-7}	$.16 \times 10^{-6}$	18,630	1285
301	3.93×10^5	69×10^3	0.5	87.6	1.417×10^{-7}	$.16 \times 10^{-6}$	14,290	986
313	7.85×10^5	138×10^3	1.0	175.3	1.417×10^{-7}	$.16 \times 10^{-6}$	17,146	1183
315	1.57×10^6	275×10^3	1.0	175.3	1.417×10^{-7}	$.16 \times 10^{-6}$	19,113	1318
319	1.57×10^6	275×10^3	1.0	175.3	1.417×10^{-5}	$.16 \times 10^{-6}$	18,953	1307
324	1.57×10^6	275×10^3	1.0	175.3	1.417×10^{-4}	160×10^{-6}	18,074	1247
321	1.57×10^6	275×10^3	1.0	175.3	1.417×10^{-3}	1.6×10^{-3}	6,800	469

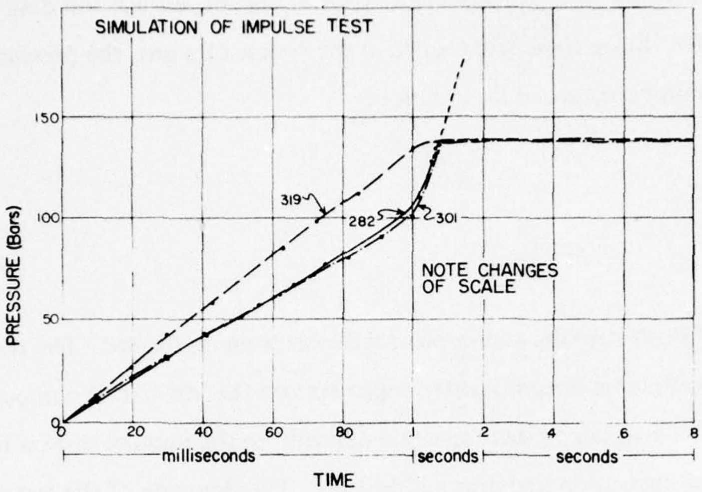
*For pressure to rise to 1950 psi.

Table 5-4 and Figs. 5-6(a) and (b) present the simulation results for the same cylinder as those for which stroking test simulation results were given in Fig. 5-3. Computer run 301 assumed that there was no relief valve in the circuit. The change in slope at approximately 0.1 seconds corresponds to the taking up of slack. It is seen that once slack is taken up the pressure rise rate is much higher, provided both the piston rod inertia and the leakage coefficient are small.

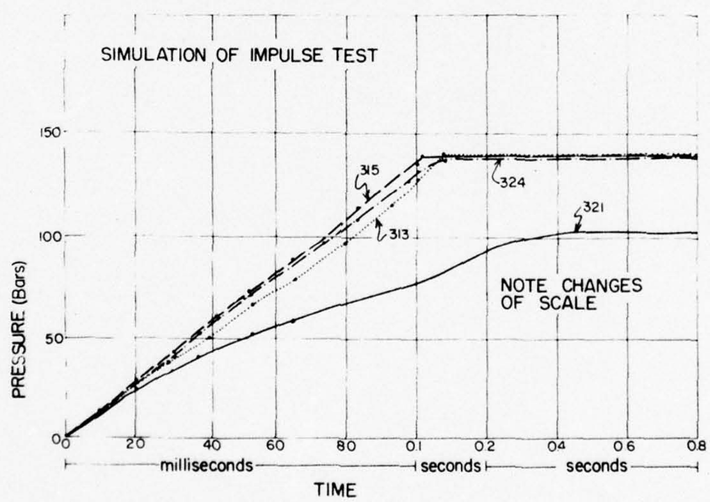
Computer run 319, and runs 324 and 321 graphically depict the degradation of the piston seals during impulse testing. In fact, when the leakage coefficient has increased a hundred fold, as in 321, not only is the pressure rise rate low, but the final pressure itself is less than the relief valve setting. This indicates that much of the pump flow is leaking past the piston rod.

COMPARISON

From the simulation runs, it can be seen that, for the same flow rate, the pressure rise is approximately twice as large for the impulse test as for the stroking test. Or, in other words, a given pressure rise rate can be achieved with a much smaller pump in an impulse test. In both tests, the pressure rise rate is significantly affected by the test system parameters (e.g.,



(a)



(b)

Fig. 5-6. Impulse Test Simulation. (Numbers refer to computer run results. See Table 5-4 for parameters.)

inertia of moving parts, relief valve characteristics, and in the case of the impulse test the frame stiffness and slack). It is also affected, however, by the leakage and drag characteristics of the cylinder. Since these will change in the course of a test, the pressure rise rate can be used to monitor the rate of degradation.

CONCLUSION

Simulation of both stroking and impulse tests has been performed. The results show that the pressure wave form is significantly dependent on the test circuit components and the testing fixture. Thus, the pressure rise rate depends in the impulse test on frame stiffness and in the stroking test on seal drag and leakage. The flow rate of the test system pump is also an influential factor.

CHAPTER VI

EXPERIMENTAL EFFORT

The main objective of the experimental effort in this phase of the U.S. Army MERDC Reliability Program was to ascertain the correlation between strain rates and pressure rise rates when hydraulic cylinders are subjected to cyclic loading. Test specifications (See Appendix A.) usually call for a certain minimum pressure rise rate to be maintained, but it should be remembered that the fatigue life of a component is really a function of strain rates. It is generally impractical to strain gage every cylinder subjected to fatigue testing; whereas, pressure measurements can be relatively easily obtained. Hence, the importance of establishing the correlation between strain rates and pressure rise rates can be seen.

Since the hoop and longitudinal stresses in a hydraulic cylinder depend only on the pressurization and not on rod velocity, it was decided to measure pressure and strain in a test setup which duplicates the locked rod test. Fig. 6-1 presents the test circuit schematic. Pressure and strain cycles can be imposed by moving the directional control valve as well as energizing and de-energizing the solenoid relief valve. The needle valve permits control of the pressure rise rates.

The test cylinder was fixed with the rod extended mid-stroke. Suitably chosen strain gages were attached to measure axial and hoop strain on opposite sides of the cylinder surface. (See. Fig. 6-2.) Strain gages were connected so as not to be affected by bending stresses due to self-weight or other reasons. By using identical strain gages on all four legs of the resistance bridges, temperature effects were automatically compensated. Table 6-1 summarizes the dynamic characterstics of the instrumentation used during the tests.

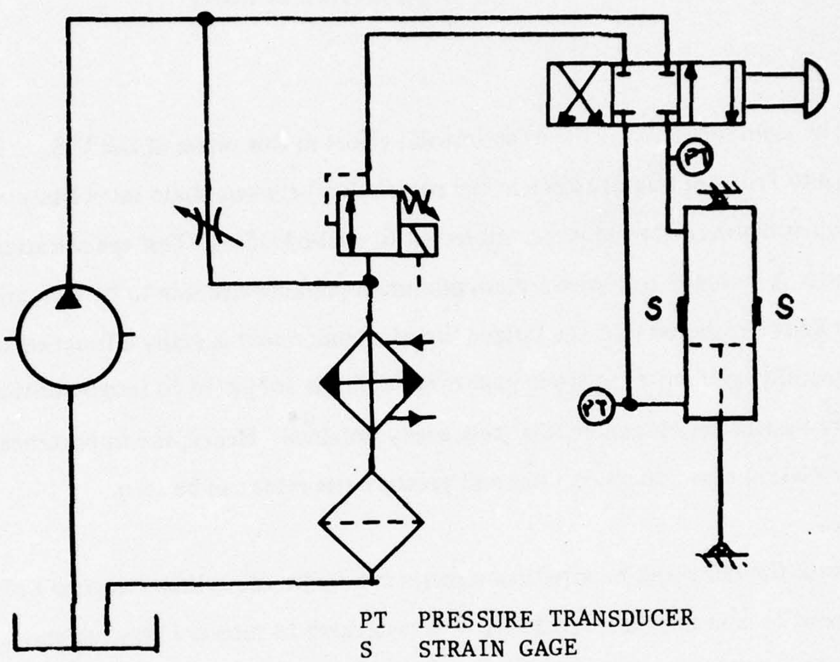


Fig. 6-1. Experimental Setup Used for Pressure Strain Correlation.

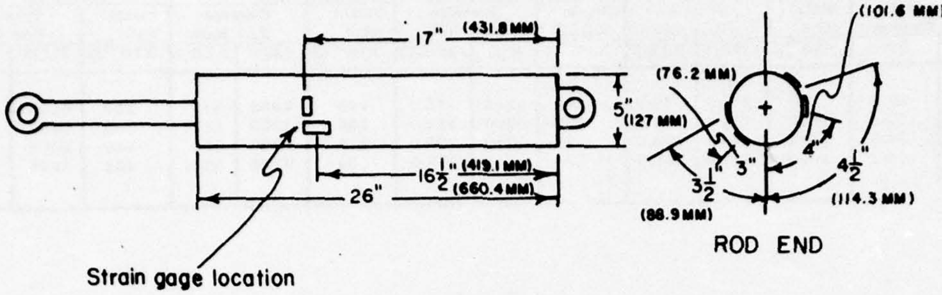


Fig. 6-2. Location of Strain Gages on Test Cylinder.

TABLE 6-1. DYNAMIC CHARACTERISTICS OF INSTRUMENTATION USED FOR LOCKED ROD TESTS.

Serial Number	Instrument	Dynamic Response
1	*Budd Strain Recorder*, Model P-350	0 – 50 Hz.
2	Bell & Howell Pressure Transducers, Strain Gage Bridge Type (0-5000 psi)	0 – 16 kHz.
3	Honeywell CRT Visicorder (recording oscillograph)	0 - 1 kHz.

Table 6-2 summarizes the strain and pressure rates for different pressure settings. Figs. 6-3, 6-4, 6-5, and 6-6 are typical oscillograph recordings for pressure cycling obtained by using the solenoid relief valve. It can be seen from the figures that the pressures and strains rise simultaneously. However, the strain decay is seen to exhibit transient spikes in Figs. 6-3 and 6-6, which have no corresponding feature in the pressure traces. From this, it can be inferred that, under certain circumstances, the material of the cylinder wall may be subjected to stress rates which are not reflected in pressure measurements. If the cylinder is made of material

TABLE 6-2. STRAIN RATES AND PRESSURE RISE RATES FOR LOCKED ROD TESTS.

Stress Measured Pressurization Final Pressure		Hoop						Axial					
		Head End			Rod End			Head End			Rod End		
		Strain Rate	Pressure Rise Rate		Strain Rate	Pressure Rise Rate		Strain Rate	Pressure Rise Rate		Strain Rate	Pressure Rise Rate	
psi	bars	$X10^{-6}/S$	PSI/S	BAR/S	$X10^{-6}/S$	PSI/S	BAR/S	$X10^{-6}/S$	PSI/S	BAR/S	$X10^{-6}/S$	PSI/S	BAR/S
1000	69.0	167	1250	86.2	140	1115	76.9	140	5000	345	250	5000	345
1500	103.5	1080	6000	414.0	120	6000	414.0	260	7500	517	400	6000	414
2000	138.0	1296	5715	394.0	90	5720	395.0	270	8000	552	540	8000	552
2500	172.4	2000	6250	431.0	135	7145	493.0	130	8335	575	455	7145	493

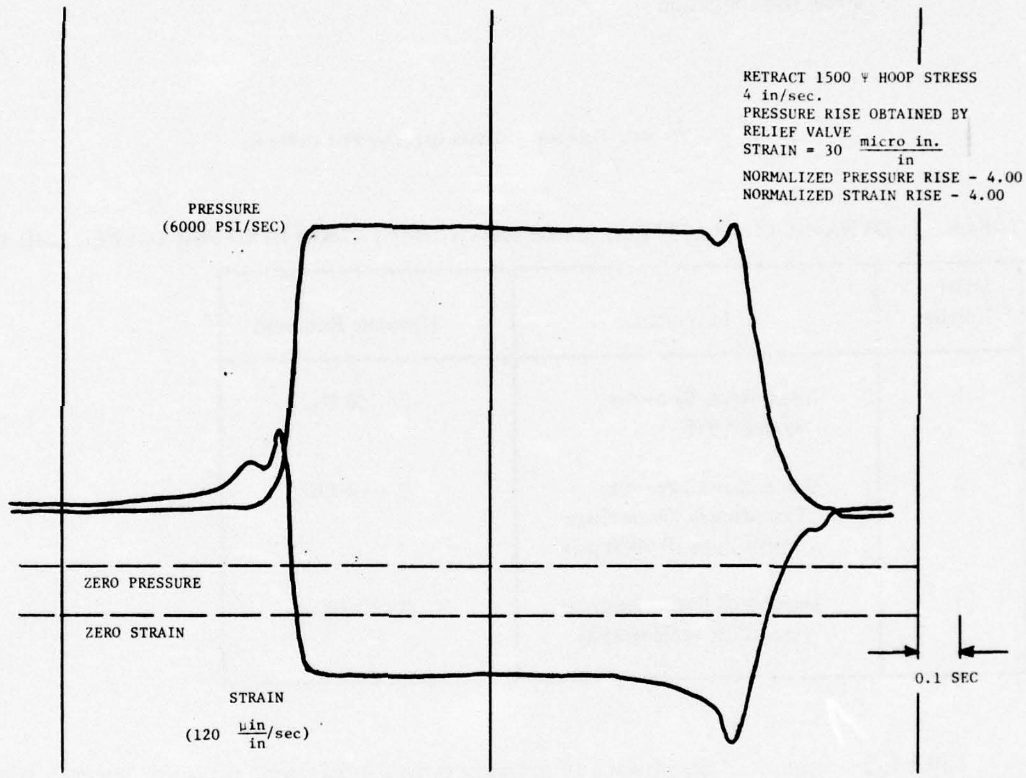


Fig. 6-3. Dynamic Pressure and Hoop Strain for a Locked Rod with Rod End Pressurized.

whose fatigue strength if not affected by the strain rate, the discrepancies between the pressure and strain rates as reported above can be ignored. If such is not the case, strain measurements must be made at least for a few cycles to insure that specified strain rates are maintained.

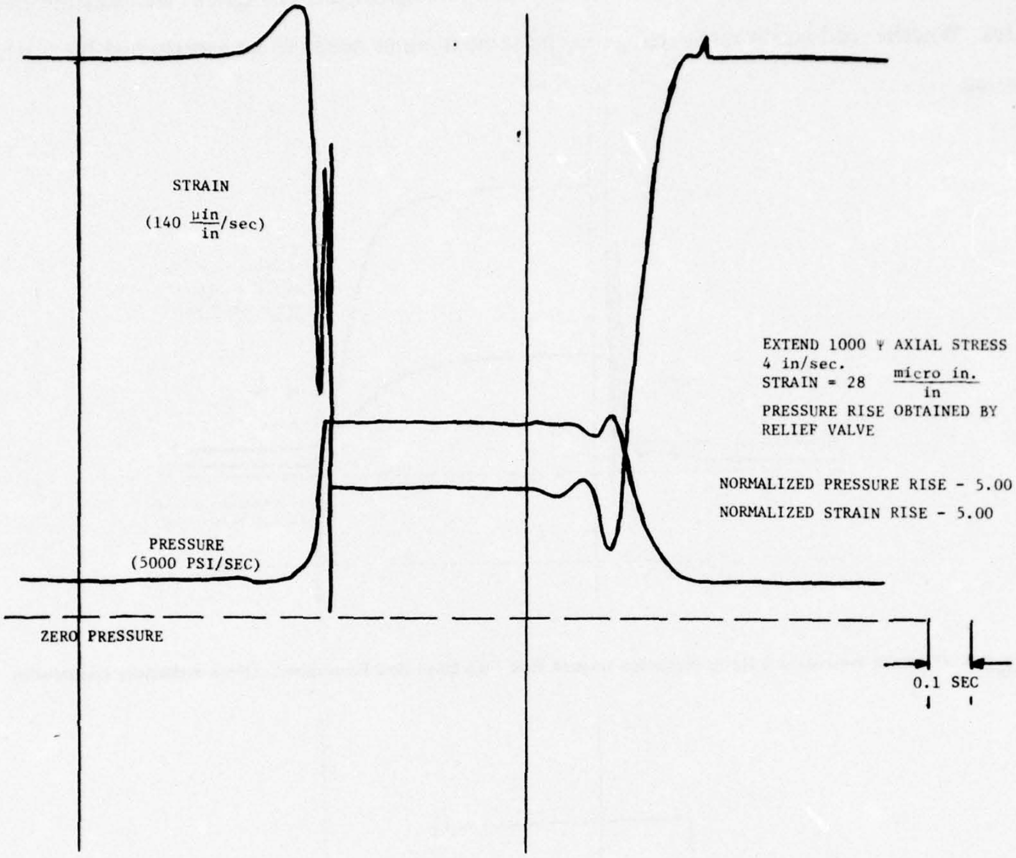


Fig. 6-4. Dynamic Pressure and Axial Strain for a Locked Rod with Head End Pressurized. (Note oscillatory transient in strain.)

It should also be noted that, if the strain has a wave form different from that of the pressure, the cycle life for strain may not be identical with the cycle life for pressure. It should also be pointed out that the correlation between cylinder strain and pressure is the same for both impulse and stroking tests. Consequently, the above inferences on strain and pressure rates are valid for both kinds of test, even though they were obtained in a test setup in which the cylinder rod was fixed at mid-stroke. Further tests are needed to examine the effect of pin-eye clearances on the strain rates for cylinder rods. Simulation of locked rod tests discussed in

Chapter V shows that pin-eye clearances can result in significant changes in the pressure rise rates. Whether rod strain rates change in the same manner needs to be ascertained by further testing.

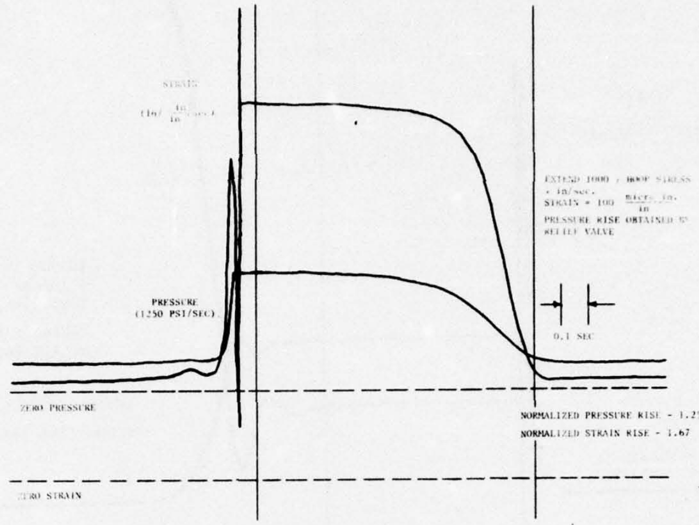


Fig. 6-5. Dynamic Pressure and Hoop Strain for Locked Rod with Head End Pressurized. (Note oscillatory transient in strain.

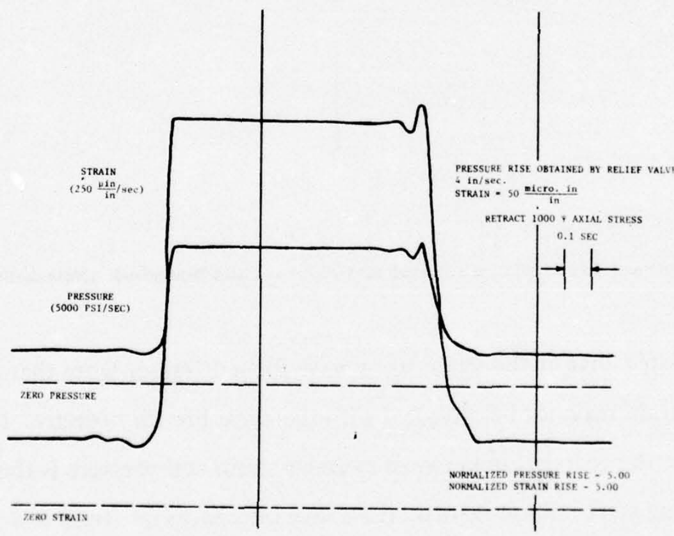


Fig. 6-6. Dynamic Pressure and Axial Strain for Locked Rod with Rod End Pressurized.

CHAPTER VII

A REVIEW OF TEST PROCEDURES OF FATIGUE LIFE EVALUATION OF HYDRAULIC CYLINDERS

This chapter has been motivated by a review of the current test procedure as well as the test data presented in numerous tests conducted according to the procedures. Appendix A contains the relevant portions of the test procedures under review and Appendix B the references for the test reports studied. It is appropriate to make a few general comments before discussing details of the procedures under scrutiny.

The prime purpose of developing a standard test procedure is to permit the objective evaluation of the suitability of a piece of hardware for the intended application. Admittedly, the best method of doing so is to install the component on the system on which it is intended to be used and subject it to normal usage. If the component fails before the desired life has been reached, it may be rejected. There are two main reasons why this method of evaluating component endurance is often impractical. First, it may require an inordinate length of time and, second, the number and magnitude of uncontrollable variables seriously affects the reliability of the test data. Consequently, both manufacturers and users of components have evolved standard testing methods for appraising the endurance of components. Even though the test conditions may not duplicate the actual working condition for a specific application, it is conjectured that the test results can, in some way or other, be related to actual field life. Fatigue life testing of components usually involves subjecting them to cyclic stresses of either a deterministic or random wave form. Since tests using random wave form cycles are harder to interpret, they will be ignored for the purposes of this discussion. In the case of hydraulic cylinders, there are so many parameters which can be varied that it is impossible to compare test results unless either (a) all test parameters are identical or (b) the influence of the test parameters which are different from test to test are precisely known. If the influence of

certain parameters is unknown or incompletely understood, it is generally preferable to insist that they be held constant for all tests which need to be compared. One of the purposes of this review is to highlight the parameters which could influence cylinder endurance testing and indicate how they can be rigorously specified.

An extremely important consideration in drafting a test procedure for fatigue testing of a component is the failure criterion. Hydraulic cylinders being heterogeneous components, there can be a number of modes of failure (e.g., seal wear out – as exhibited by leakage, rod breakage, clevis breakage, plastic distortion of eyes, etc.). Some of these (e.g., seal wear out) may be considered repairable, and the test may be interrupted to carry out such repairs as and when needed. If such is the case, it should be explicitly indicated in the test procedure. No other type of repair or reconditioning on the test cylinder should be permitted. Sometimes, however, the proper completion of a fatigue test on a component requires the repair/conditioning of other elements in the test fixture or circuit. Thus, in testing hydraulic cylinders, it may be necessary to replace deformed or broken pins. Such repairs should be permitted, since otherwise it can be contended that the existence of such defective components led to the premature failure of the cylinder itself. It should be noted, however, that the defective elements (e.g., pins) should be changed as soon as defects are noted. Stress distribution in the test cylinder parts may be radically altered if such defective elements are allowed to stay till breakage occurs.

In addition to the above general remarks, the following specific comments are offered for consideration during revision of the test procedure:

1. **Paragraph 4.5.2.2 is the only reference to the test stand. Only the test circuits for packing drag and endurance tests are indicated. The test circuit for impulse endurance is not specified, and one would reasonably assume that either of the test circuits furnished could be used for such a test. However, neither of the test circuits shown in Fig. A-1 or A-2 have provisions for positively holding the cylinder rod. Consequently, an impulse test conducted by using one of the above two circuits would not be comparable to one in which the cylinder rod was mechanically locked in mid-position. (See Fig. 3-1.) Various test reports studied for historical data (See Appendix B.) indicate that impulse tests were performed using a frame similar to Fig. 3-1.**

Structural features of the test fixture have a significant influence on the stress cycles to which various parts of the test cylinder are subjected. Simulation results presented in Chapter V of this report show, for example, how the frame stiffness, inertia of moving parts, and packing drag affect the stress rise rates. In addition to these parameters, the eccentricity and angular misalignment of the mountings in the test fixture can seriously affect the stress levels in the test cylinder and rod. The fit between the pins and the eyes also affects local stress levels. Unless all of the above parameters are controlled, it is difficult to compare test results obtained on different cylinders and especially on different test stands.

2. **Paragraph 4.5.2.2.4 refers to *"rated full flow."* Commercial cylinders are rarely assigned a rated flow in view of the fact that the flow rate depends on the specific application. It would appear more reasonable to specify the displacement wave form to accompany Fig. A-3.**

As indicated in Chapter III, the pressure wave form will approximate a trapezoid, and the tolerances on this wave form should be specified. Since the pressure rise rate is zero at the beginning and at the crest of a trapezoidal wave form, the stipulation *"minimum rate of pressure rise shall be 20,000 psi per second"* is, strictly speaking, unattainable.

"Malfunction" criteria should be described explicitly. As indicated in the general remarks, seals and pins may require replacement in the course of a test. It is presumed that the objective of the test is to appraise the cylinder structure and not the seals or pins. However, repairs to tie rods, end caps, and other components which are normally part of the structural unit should not be permitted.

3. **The description of the impulse endurance test, as given in Paragraph 4.5.2.2.5, does not specify the test fixture to be used. It is preferable to describe the wave form to be used rather than to indicate a *"pressure rise rate."* Reference to *"full flow"* may also be thereby avoided.**

Mention should be made of the efforts of professional societies in standardizing apprisal techniques for fluid power components. Personnel of the Fluid Power Research Center are

active participants in deliberations of the B93 Ad Hoc Committee of the American National Standards Institute, dealing with pressure rating of fluid power components. Since some of the major technical issues under discussion have not been resolved, it is difficult to visualize the impact on standard test procedures. The National Fluid Power Association has issued two draft recommended standards for pressure rating of hydraulic cylinders [Refs. 10 and 11]. These standards deal, however, only with the pressure containing envelope and will need to be appropriately supplemented to allow for failure in other parts of a hydraulic cylinder.

In conclusion, it must be said that the cylinder evaluation test procedures, as currently drafted, do not permit the comparison of test results from different facilities for the following reasons:

1. **Test system parameters are not completely specified. Simulation data presented in Chapters IV and V show how stress rise rates can be affected by test system parameters.**

2. **Failure criteria are not explicitly indicated. Comparison of stroking and impulse tests, for example, is meaningless unless the same failure mode is specified.**

3. **Statistical aspects of testing and data interpretation are completely absent. This gap is serious in that fatigue failure is essentially a random phenomenon and can be realistically described only in probabilistic terms (i.e., using means, variances, confidence intervals, sample sizes, and the like). The drawing of conclusions from tests on single samples is especially fraught with danger, since fatigue life of metallic parts can have a scatter extending over an order of magnitude. The remendy is primarily twofold: first, identify and control the test parameters as closely as possible – this will reduce data scatter; second, test at least five components to failure. (Five is certainly an arbitrary figure, but the information gathered from any less would exhibit sample bias seriously.)**

Tests on heterogeneous components, such as hydraulic cylinders, should be mainly used to validate assembly design rather than material properties. Possible modes of failure are more varied for assemblies than for material testing specimens, and test reports should be thoroughly documented so that data are not misused by comparison with different failure modes.

Since tests on complete assemblies (e.g., hydraulic cylinders) are generally of longer duration than test coupons, the inclination to terminate tests once a number of cycles has been reached is understandable. However, such test data are of no use in building up a statistical population on which to base confidence intervals, test sample sizes, etc. Population sizes for assembly testing are intrinsically small, and it is felt that every effort should be made to enlarge

them. In the case of hydraulic cylinders, this can be done by testing cylinders to destruction rather than discontinuing after 1,500,000 cycles or so.

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APPENDIX A

EXCERPTS FROM

(PROPOSED)

MILITARY SPECIFICATION

CYLINDER, HYDRAULIC, DOUBLE-ACTING

2000 PSI (MAXIMUM)

EXCERPTS FROM

(PROPOSED)

MILITARY SPECIFICATION

CYLINDER, HYDRAULIC, DOUBLE-ACTING

2000 PSI (MAXIMUM)

1. SCOPE

1.1 *Scope* This specification covers the 2000 psi double-acting hydraulic cylinders for use on stationary and mobile equipment.

1.2 *Classification* Hydraulic cylinders shall be of the following types as specified:

TYPE I	**1,500,000 Duty Cycles (Heavy Duty)**
TYPE II	**750,000 Duty Cycles (Moderate Duty)**
TYPE III	**60,000 Duty Cycles (Light Duty)**

3.5 *Performance* The cylinder shall operate at 2000 psi at all flows up to rated flow. The cylinder shall perform as specified herein without buckling, bending, or leakage.

3.5.1 *Proof Pressure* The cylinder shall withstand a minimum proof pressure of 4000 psi when tested as specified in 4.5.2.2.1 without permanent deformation, damage, or leakage.

3.5.2 *Piston Drift* The piston shall drift a maximum of 1 inch per hour when supporting a load generating 2000 psi hydraulic pressure at an oil temperature of 150° F plus or minus 5° F.

3.5.3 *Rod Seal Leakage* Rod seal leakage shall not exceed 1/2 cc per hour at full stroke at an oil temperature of 150° F plus or minus 5° F and a pressure of 2000 psi.

3.5.4 *Piston and Rod Drag* The combined drag shall not exceed 3 psi per inch or bore diameter for the rod end and 2 psi per inch of bore diameter for the head end when tested as specified in 4.5.2.2.3.

3.5.5 *Cyclic Endurance* The cylinder shall meet the performance requirements as specified herein and shall show no evidence of leakage or damage after having been operated, as specified in 4.5.2.2.4, for one-half the number of cycles of each type of cylinder.

3.5.6 *Impulse Endurance* The cylinder shall meet the performance requirements as specified herein and show no evidence of leakage or damage after having been operated, as specified in 4.5.2.2.5, for one-half the number of cycles of each type of cylinder.

4.5.2.1.3 *Test Apparatus* All test apparatus shall be accurate within limits specified in Mil-Std-448, Section 6.

4.5.2.2 *Test Procedure* Typical test circuit is shown in Figs. A-1 and A-2.

4.5.2.2.1 *Proof Pressure* Position and mechanically hold the piston at mid-point of cylinder. Fill both sides of piston with oil. With the head end port capped, apply an oil pressure of 4000 psi to rod end of the piston for 30 seconds. With the rod end port capped, apply an oil pressure of 4000 psi to head end of the piston for 30 seconds. Any evidence of external leakage or deformation shall constitute failure of this test.

4.5.2.2.2 *Piston Drift* Position and hold the piston of the cylinder at mid-point of the cylinder. Fill both sides of cylinder with oil. Cap the head end port and vent the rod end port to atmosphere. Pressurize the head end of the slave cylinder to attain a minimum of 2000 psi at the head end of the test cylinder. Maintain this pressure for 15 minutes. Measure and record the travel of the piston during the 15 minutes. Repeat above test capping rod end port, venting the head end port to atmosphere, and pressurizing the rod end of the slave cylinder. Piston drift greater than one inch per hour shall constitute failure of this test.

4.5.2.2.3 *Packing Drag* Position the piston at the mid-point of the cylinder. Fill both sides of the cylinder with oil and vent the head end to atmosphere. Gradually pressurize the rod end of the cylinder. Record the minimum pressure at which the piston moves and also the pressure required to keep it in motion. Repeat the above test by pressurizing the head end of the cylinder. Head end pressure exceeding 3 psi per inch bore diameter or rod end pressure exceeding 2 psi per bore diameter shall constitute failure of this test.

4.5.2.2.4 *Cyclic Endurance* Cycle the test cylinder in accordance with the schedule shown in Fig. A-3 at rated full flow and at maximum operating pressure for one-half the number of cycles based on classification of cylinder (Sec. 1.2). Oil temperature shall be 180° F plus or minus 5° F during the balance of test run. The minimum rate of pressure rise shall be 20,000 psi per second. Upon completion of above cycling, repeat the piston drift (4.5.2.2.2) and packing drag (4.5.2.2.3) tests specified herein. Malfunction prior to completion of specified cycles, failure to meet the criteria set forth in the above specified tests, or evidence of external leakage or damage shall constitute failure of this test.

4.5.2.2.5 *Impulse Endurance* Position the piston rod extended halfway in the test cylinder and pressurize alternately the cylinder ports at a rate of 30 cycles per minute with a pressure rise rate greater than 20,000 psi per second. Cycle the test cylinder at full flow and at maximum operating pressure for one-half the number of cycles based on the classification (Sec. 1.2). Upon the completion of the above cycling, repeat the piston drift and packing drag tests specified herein. Malfunction prior to completion of specified cycles, failure to meet the criteria set forth in the above specified tests, or evidence of external leakage or damage shall constitute failure of this test.

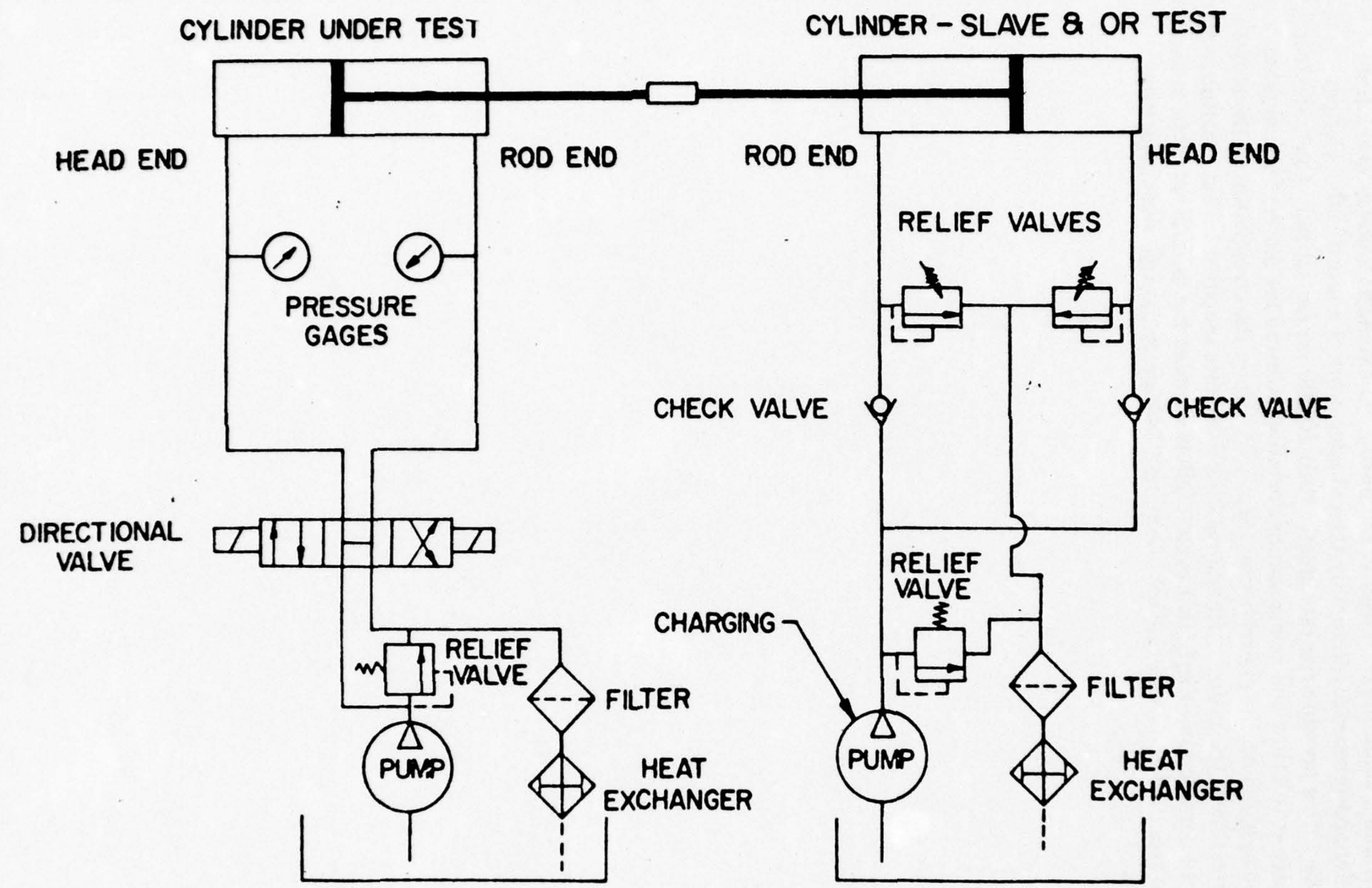

FIGURE A-1 TYPICAL TEST SETUP FOR ENDURANCE TESTING.

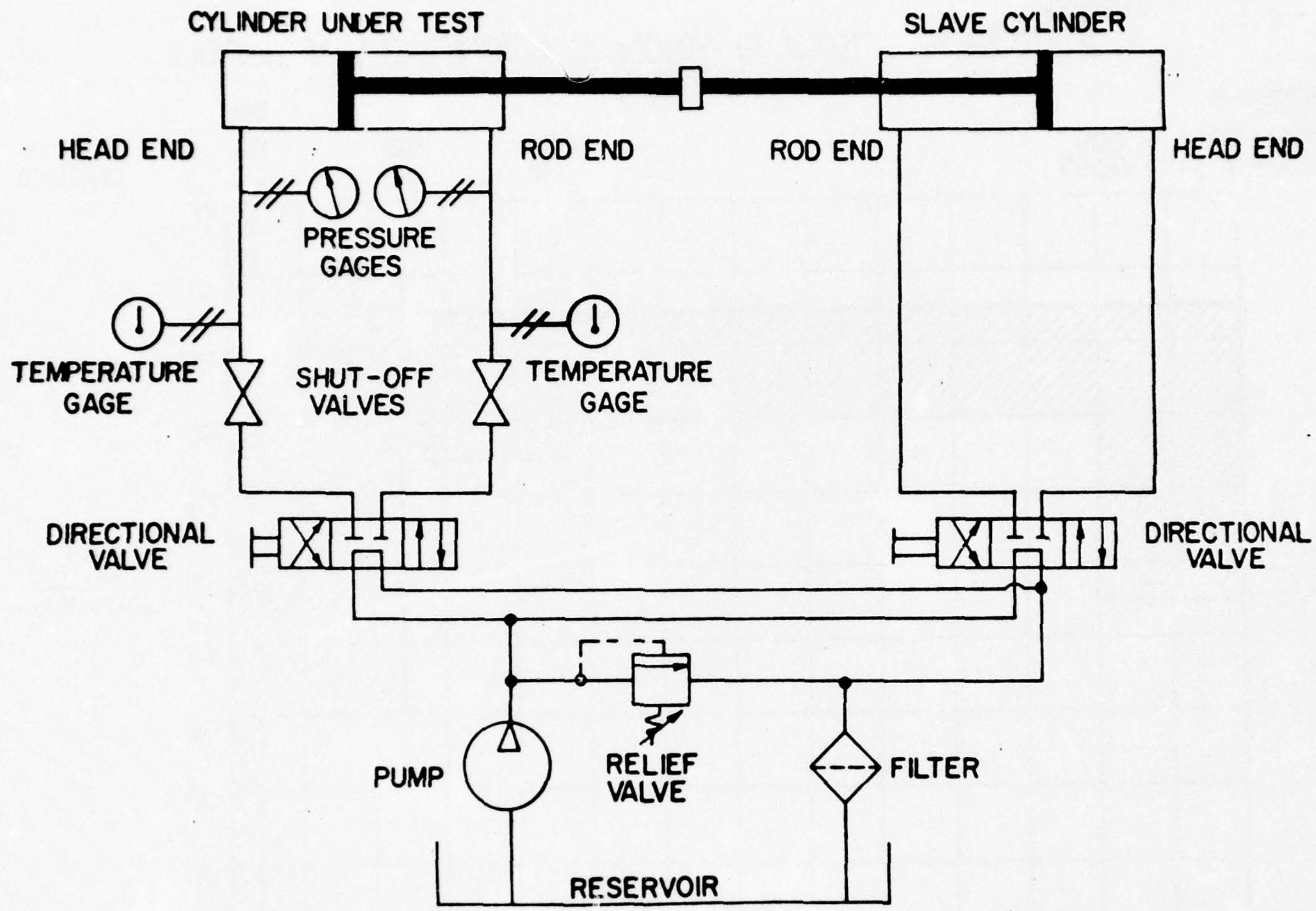

FIGURE A-2 TYPICAL TEST SETUP FOR LEAKAGE AND PACKING DRAG DETERMINATIONS

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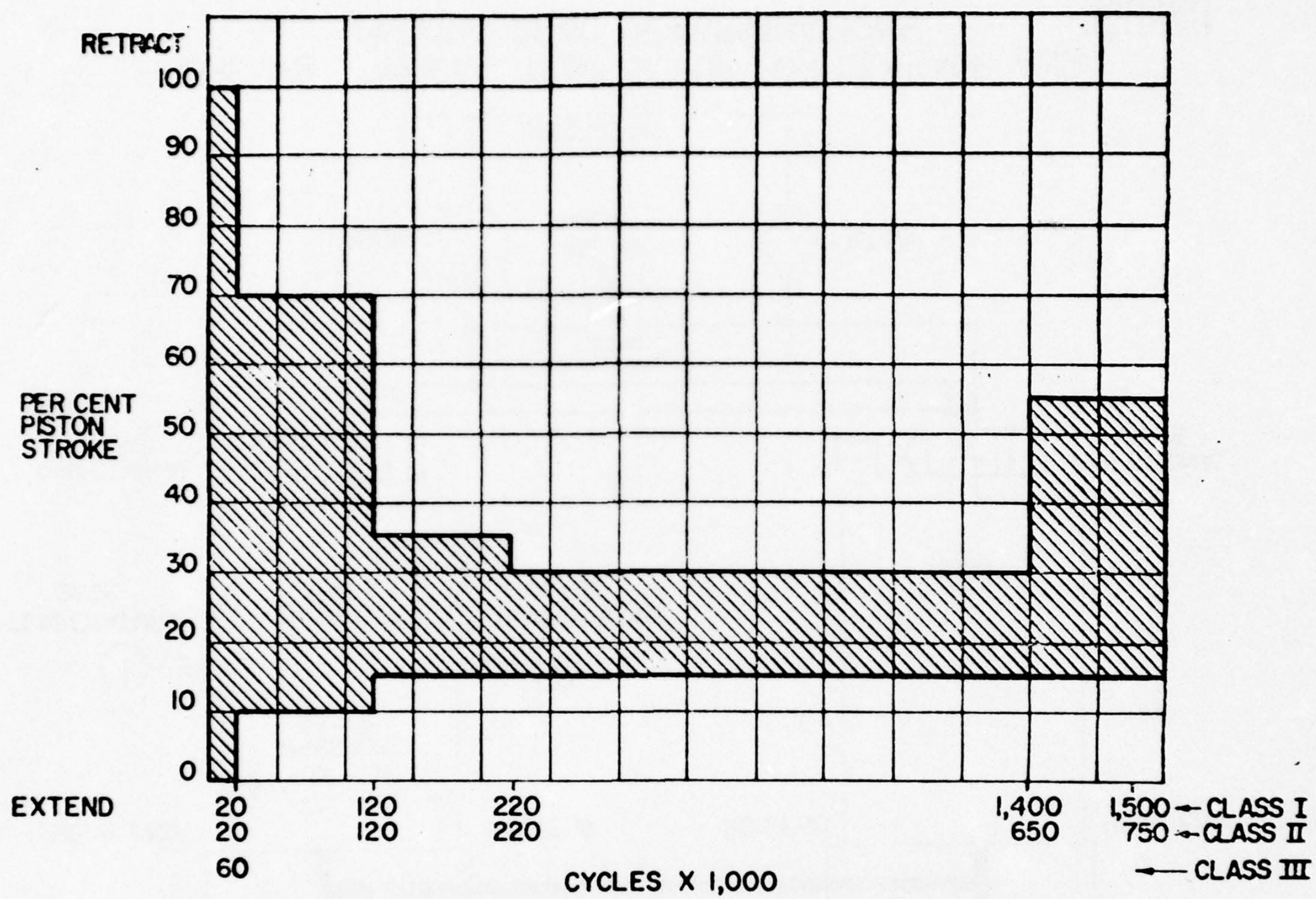

FIGURE A-3 CYLINDER STROKE & CYCLE SCHEDULE

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APPENDIX B

TEST REPORTS ON HYDRAULIC CYLINDER

ENDURANCE AND IMPULSE TESTING

TEST REPORTS ON HYDRAULIC CYLINDER

ENDURANCE AND IMPULSE TESTING

TEST REPORT NO.	OSU NO.	TITLE
325 324	1000	REPORT OF EVALUATION TEST OF: HYDRAULIC CYLINDERS; Volume 1; Caterpillar & International Harvester; 3 Copies
251	1001	CYLINDERS; Volume 1; Anthony Company, Inc.; Streator, Illinois; 2 Copies
291	1002	FABRICATION & TESTING OF CYLINDERS; Volume 1; Clark Equipment Co.; Buchanan, Michigan; 1 Copy
264	1003	CYLINDER REPORT; Volume 1; Hyco, Inc.; Ashland, Ohio; 1 Copy
244A	1004	CYLINDERS; Volume 1; Frank G. Hough Co.; Libertyville, Illinois; 1 Copy
242	1005	CYLINDERS; Volume 1, Caterpillar Tractor Co.; Peoria, Illinois; 1 Copy
266	1006	CYLINDERS; Volume 1 and Volume II; Euclid – Division of General Motors Corporation; Hudson, Ohio; 1 Copy
273	1007	CYLINDERS; Volume 1; Allis-Chalmers; Milwaukee, Wisconsin; 1 Copy
293A	1008	CYLINDERS; Volume 1; Miller Fluid Power – Division of Flick-Reedy Corporation; Bensenville, Illinois; 1 Copy
302	1009	CYLINDERS; Volume 1; Parker-Hannifin Corporation, Des Plaines, Illinois; 1 Copy
243	1010	CYLINDERS; Volume 1; Cascade Manufacturing Co.; Portland, Oregon; 1 Copy

TEST REPORT NO.	OSU NO.	TITLE
293	1011	CYLINDERS INTERCHANGEABILITY; Volume 1; Miller Fluid Power; Division of Flick-Reedy Corporation; Bensenville, Illinois; 1 Copy
321	1012	TELESCOPIC CYLINDERS; Volume 1; 1 Copy
244A	1013	SUPPLEMENT TO REPORT OF TESTING OF HOUGH HYDRAULIC CYLINDERS; Volume 1; 1 Copy
272	1014	FILTERS; Volume 1; Marvel Engineering Co.; Chicago, Illinois; 1 Copy
270D	1015	20 TON ROUGH TERRAIN (PES) CRANE; American Hoist & Derrick Company; St. Paul, Minnesota; 1 Copy
270B	1016	STEERING CYLINDER CONTAMINATION FOR 20 TON ROUGH TERRAIN CRANE; Volumes 1, II, and III; American Hoist & Derrick Company; Fort Wayne, Indiana; 1 Copy
270C	1017	CYLINDER MOUNTING EYE FAILURE, 20 TON ROUGH TERRAIN CRANE; Volumes 1 and II; American Hoist and Derrick Company; Fort Wayne, Indiana; 1 Copy
270K	1018	FAILURE OF THE SPIROLOX RETAINING RING IN CLARK CENTERING CYLINDER FOR 20 TON ROUGH TERRAIN CRANE; Volume 1; Clark Equipment Company; Buchanan, Michigan; 1 Copy
270A	1019	RETAINING RING FAILURE FOR 20 TON ROUGH TERRAIN CRANE; Volume 1; Ramsey Corporation; St. Louis, Missouri; 1 Copy
270L	1020	EVALUATION OF PROTECTIVE BOOTS FOR 20 TON ROUGH TERRAIN CRANE; Volume 1; 1 Copy
270M	1021	EVALUATION OF COMPONENTS FOR HYDRAULIC CONTROL CYLINDER, 20 TON ROUGH TERRAIN CRANE; Volume 1; American Hoist & Derrick Company; Fort Wayne, Indiana; 1 Copy

TEST REPORT NO.	OSU NO.	TITLE
DTB05R74	1022	HYDRAULIC CYLINDER, LIMITED QUALIFICATION TEST OF; Prince Manufacturing Corporation; Sioux City, Iowa; 1 Copy
DTB05R74	1023	HYDRAULIC CYLINDER, LIMITED QUALIFICATION TEST OF; Tomkins-Johnson Company; Jackson, Michigan; 1 Copy

Reports 1022 and 1023 done by Dayton T. Brown Testing Laboratories, Inc. All others done by Allied Research Associates.

SECTION II

HYDRAULIC NOISE

PROJECT STAFF

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FOREWORD

This section presents a detailed account of the project activities in the area of hydraulic noise. Specific test procedures are recommended for determining the fluidborne noise generation potential of hydraulic pumps and the acoustical performance characteristics of fluidborne noise attenuators. Experimental data are shown to indicate typical results that are obtained with the proposed test codes.

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CHAPTER I

INTRODUCTION

Three types of noise occur in fluid power systems – fluidborne noise, structureborne noise, and airborne noise. Hydraulic components generally generate, transmit, or radiate one or more of these noise forms. The control of fluid power system noise requires assessing the acoustical performance of system components and using the resulting information to construct the system in an optimal fashion to achieve the quietest possible configuration.

Table 1-1 [1] shows the acoustical characteristics of hydraulic components which should be assessed and reported to a system designer to allow the proper acoustical design of fluid power systems. The *"stars"* in Table 1-1 indicate those assessments that are considered to be the most important. As indicated in the table, test codes are available for measuring the airborne noise emitted by pumps and motors. The objective of this study is the development of industrially acceptable test procedures for evaluating: (1) the fluidborne noise generation characteristics of hydraulic pumps, and (2) the fluidborne noise reduction characteristics of pressure ripple attenuators.

The project plan of attack for the reporting year was:

1. **Obtain data to assist in the development of a procedure for measuring the fluidborne noise generation potential of a hydraulic pump. Transmit that data to appropriate pump/motor standards committees of NFPA.**
2. **Prepare a test procedure for determining the effectiveness of fluidborne noise attenuators. Obtain data using the procedure.**

Implementation of this plan produced excellent progress in both of the target areas. The following chapters delineate the results of this year's study of both pumps and fluidborne noise attenuators, present two recommended procedures, and show specific test results obtained with the recommended procedures.

TABLE 1-1. REQUISITE ACOUSTICAL CONSIDERATIONS FOR ESTABLISHING FLUID POWER COMPONENT EVALUATIONS [1].

HYDRAULIC COMPONENT	NOISE CATEGORY		
	FLUIDBORNE	STRUCTUREBORNE	AIRBORNE
PUMP	☆ GENERATION	GENERATION TRANSMISSION	☆ EMISSION test code
CONDUIT	TRANSMISSION	TRANSMISSION	EMISSION
VALVE	GENERATION TRANSMISSION	GENERATION TRANSMISSION	EMISSION
MOTOR	GENERATION	GENERATION TRANSMISSION	☆ EMISSION test code
FLUIDBORNE NOISE CONTROLLERS	☆ ABSORPTION REACTION	TRANSMISSION	—
STRUCTUREBORNE NOISE CONTROLLERS	—	ABSORPTION REACTION	—
AIRBORNE NOISE CONTROLLERS	—	TRANSMISSION	ABSORPTION REACTION

CHAPTER II

PUMP FLUIDBORNE NOISE GENERATION POTENTIAL

The fluidborne noise generated by fluid power pumps has been discussed in several papers over the past five years [2, 3, 4, 5]. Ichikawa and Yamaguchi [2] developed a model for the variation in flow and pressure (FBN) generated by a gear pump. From the data given, the model proved satisfactory for predicting the flow ripple of the gear pump. They consider the leakage (or source) impedance in calculations to determine the pressure ripple delivered to the line.

Willikens [3] presents a model for a gear pump and presents data to support his model. He also shows that the pressure ripple measured is dependent on both the flow ripple generated by the pump and the leakage impedance of the pump.

References [4] and [5] both consider ways of rating and measuring the flow ripple of hydraulic pumps. Ref. [4] attempts to make a case for rating pumps on the basis of pressure ripple for a zero length outlet condition. This is accomplished by placing a valve next to the pump, within 1/20 of the wave length of interest, and installing a pressure transducer between the pump outlet and the valve.

Unruh [5, p. 2] recognizes the need to ultimately relate the measured pressure ripple back to the flow ripple of the pump. He states "*... the basic phenomenon which actually needs to be established for determining the fluidborne noise characteristics of a pump is the amount of flow ripple that a pump generates due to its inherent construction characteristics ...*"

A pump is analogous to a constant current source in an electrical circuit. The pump will deliver a constant flow ripple to the system at a given operating condition. Just as the load on a constant current source determines the voltage in the circuit, so the load conditions on a fluid power pump will determine the magnitude of the pressure ripple in the system. One other similarity between a constant current source and a pump is that both have some internal or source impedance. For a pump, the source impedance will depend on the geometry (or volume) of the pump casing and the amount of leakage flow [2, 3].

The two important parameters in defining an electrical current source are the strength of current and the source internal impedance. Similarly, for a hydraulic pump, the inherent flow ripple of the pump and the source impedance are needed. A general approach for evaluating these parameters of fluid power pumps is discussed in the next section.

PROPOSED TEST PROCEDURE

There are two basic environments in which a hydraulic pump can be tested for fluidborne noise. These are anechoic and reverberant environments.

An anechoic environment would be an environment in which, ideally, there would be no reflections. The noise in the fluid would propagate down the tube and not be reflected. Since fluid power pumps have a wide range of pumping frequencies, an anechoic termination needs to be *"non-reflective"* over a wide range of frequencies.

The difficulty in obtaining a true, non-reflective termination has been discussed by Heymann [6]. He states ...*"It is ... difficult to obtain a practical reflection-free termination in liquid systems, most approaches realize at best a termination in which reflections near the filter are weak..."*

The basic concept of measuring pump pressure ripple in an anechoic environment is shown in Fig. 2-1. In Fig. 2-1, a pump is connected to an anechoic termination ahead of the system load valve. If there are no reflections at the termination, then the pressure at any point in the hydraulic line will be the same, since there will be no standing waves.

Flow Ripple

For the case shown in Fig. 2-1, the impedance at any point in the line is just the characteristic impedance, Z_o. Thus, the flow ripple from the pump is related to the pressure as:

$$\tilde{Q} = \tilde{P}/Z_o \tag{2-1}$$

where:

$\tilde{P}$	=	measured pressure ripple	
Z_o	=	characteristic impedance of the line	$= \rho C/S$
ρ	=	density of the fluid	
C	=	sonic velocity	
S	=	cross-sectional flow area	

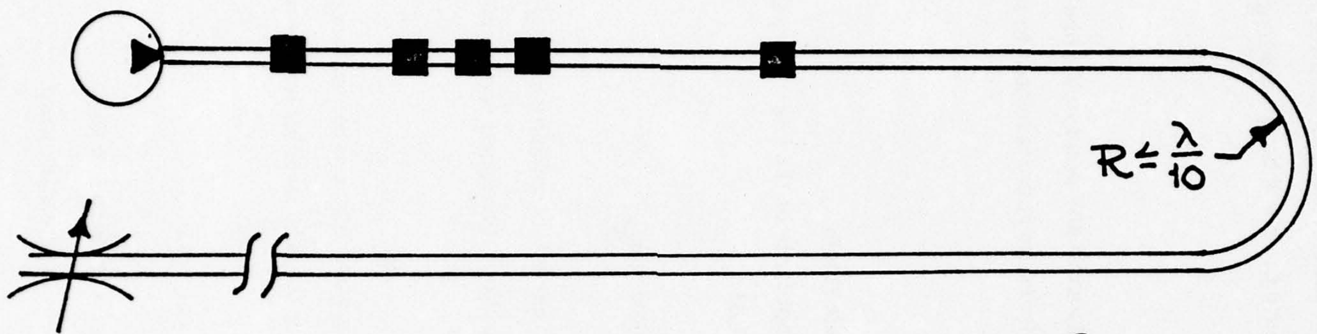
However, as mentioned previously, a truly anechoic termination can only be approached in liquid systems. Since there will be small reflections, the easiest way to measure the incident pressure is by the use of multiple transducers.

With multiple transducers [7], the standing wave ratio can be measured and the reflection factor of the termination deduced. (See Fig. 2.) The standing wave ratio is given by:

$$SWR = P_{max}/P_{min} \tag{2-2}$$

where:

P_{max}	=	maximum of the standing wave
P_{min}	=	minimum of the standing wave



$$\tilde{P}(x) = P_i \qquad Z_o = \frac{\rho c}{S}$$

$$\tilde{Q} = \tilde{P} / Z_o$$

Fig. 2-1. Basic Anechoic Test System and Equations for Evaluating the Flow Ripple of a Fluid Power Pump.

The magnitude of the reflection factor of the termination is:

$$R = |\rho_t| = SWR - 1 / SWR + 1 \tag{2-3}$$

In order to be classified as an *"anechoic"* termination for engineering purposes, the value of the reflection factor should not exceed 0.23. This corresponds to a standing wave ratio of 1.6 or approximately 4 dB.

With the reflection factor known, the value of the incident pressure ripple can be determined. This is given by:

$$P_i = P_{max} / (1 + R) \tag{2-4}$$

From the incident pressure, P_i, the value of the flow ripple from the pump can be determined with Eq. (2-1).

One approach to constructing an anechoic termination is to use very long lengths of line so that the pressure ripple is eventually dissipated as it travels down the line. For pipe diameters of one inch and flows above 3.5 gpm, a hydraulic circuit line length of between 100-200 feet is required. For larger diameters, the length required will be longer, since the frictional dissipation will decrease. Also, for larger flows, the length of line required will decrease as the flow increases. Generally, as the frequency increases, the length required should decrease. The dissipation term is thus:

$$D = F(d, Q, f) \tag{2-5}$$

where:

D	=	dissipation term
d	=	pipe diameter
Q	=	flow
f	=	frequency

Using an anechoic termination, the flow ripple from the pump can be determined. However, to evaluate the impedance of the pump, another test must be performed in which a known impedance load is inserted in the circuit similar to Fig. 2-1.

For Fig. 2-1, the pressure at any point, d (from the load), in the circuit is given by:

$$P(d) = \frac{P_o e^{-\gamma L}}{(1 - \rho_s \rho_t e^{-2\gamma L})} (e^{\gamma d} + \rho_t e^{-\gamma d}) \tag{2-6}$$

where:

P_o	=	pressure that would occur in an anechoicly terminated system with the same characteristic line impedance
ρ_t	=	termination reflection factor
L	=	distance from pump to the load
ρ_s	=	source reflection factor
γ	=	propagation coefficient

Impedance

Now the pressure, P(d), can be measured at various points in the system. P_o was measured using the anechoic termination. ρ_t is known as well as L, γ, and d. Thus, Eq. (6) can be rewritten to solve for ρ_s:

$$\rho_s = (1 - \frac{P_o e^{-\gamma L}}{P(d)} (e^{\gamma d} + \rho_t e^{-\gamma d})) \frac{e^{2\gamma L}}{\rho_t} \tag{2-7}$$

So,both the flow ripple and source impedance of the pump can be determined by semi-empirical techniques. The source reflection factor, ρ_s, can be converted to the source impedance with the following equation:

$$Z_s = Z_o (1 + \rho_s / 1 - \rho_s) \tag{2-8}$$

where: Z_s = source impedance

Z_o = characteristic line impedance

TEST RESULTS

The procedure outlined in this report for measuring the flow ripple of a hydraulic pump requires the use of an anechoic termination. This section presents the results of an evaluation of an anechoic termination and the results of pump flow ripple measurements using the qualified termination.

Anechoic Termination

Plotted in Fig. 2-2 is the reflection factor for the anechoic termination as a function of frequency. Pressure measurements were taken with six stationary pressure transducers for the two different configurations.

The largest experimental value for R was 0.176 at 167 Hz. In all cases, the value of R was below the maximum allowable value. The reflection factor is just an indication of the amount of the initial pressure wave reflected back upstream. In terms of power, a maximum of 3.1% was reflected.

The reflection factor of the anechoic termination would decrease for longer lengths of line. In such a case, the frictional dissipation would increase, thus decreasing the amplitude of a reflected wave.

The particular anechoic termination used for these tests was only evaluated at frequencies up to 400 Hz. Since the frictional dissipation term is a function of frequency, the reflection

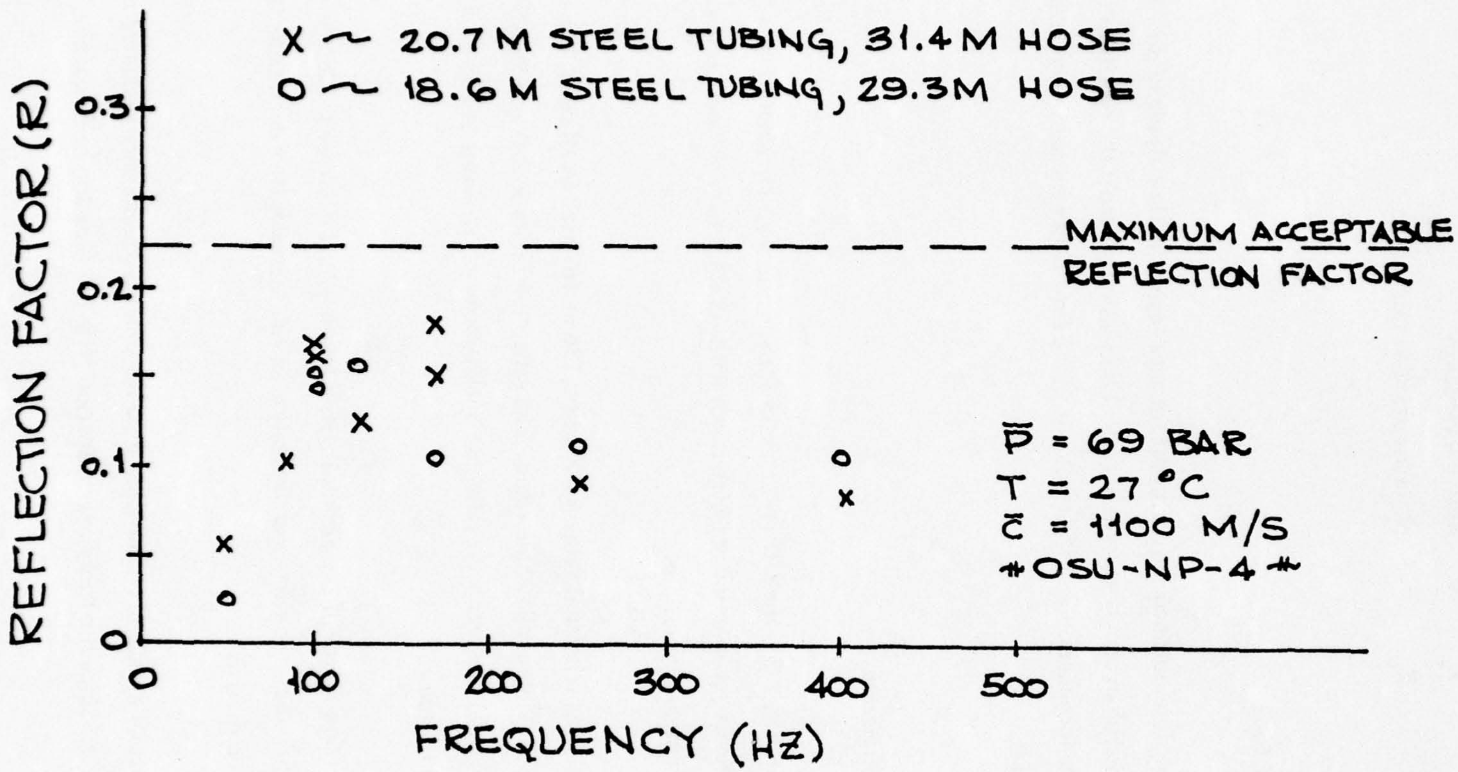


Fig. 2-2. Reflection Factor for Two Anechoic Termination Configurations Showing the Recommended Maximum Acceptable Reflection Based on 4 dB Standing Wave.

factor should decrease as the frequency increases. The molecular dissipation of acoustic waves also increases as a function of frequency.

At this point, it may seem that two contradictory things are occurring. One, we are wanting to measure the pressure ripple in a line and assuming no friction. Two, we are depending on friction to dampen the incident wave enough so that the reflection factor will be small. The measurement position should be as close as possible to the pump so that the effect of dissipation from the pump to the transducers is minimal. It is assumed that the transducer section of pipe is relatively short compared to the overall length of the line needed to obtain an anechoic termination. Thus, the frictional loss can be assumed small compared to the whole length of the system. Frictional dissipation is used to its fullest by dissipating the wave after it has passed the test circuit; thus, when it reflects, the reflection is dissipated to such an extent that, by the time it reaches the measurement section, it is small compared to the incident wave from the pump.

Pump Flow Ripple

Fig. 2-3 shows a plot of the RMS flow ripple versus speed of Pump OSU-NP-4 for the first two harmonics. The flow ripple is zero at zero RPM and, theoretically, should increase linearly as the speed of a gear pump is increased. The theoretical value of the pump flow ripple was calculated from the equation given by Ichikawa and Yamaguchi [2]:

$$q_m = a_m \cos mwt \tag{2-9}$$

where:

$a_m = 4\,b\,R_g^2\,w_o / m^2\,n$

b = gear face width

m = harmonic number

n = number of teeth

$w_o = 2\pi N$

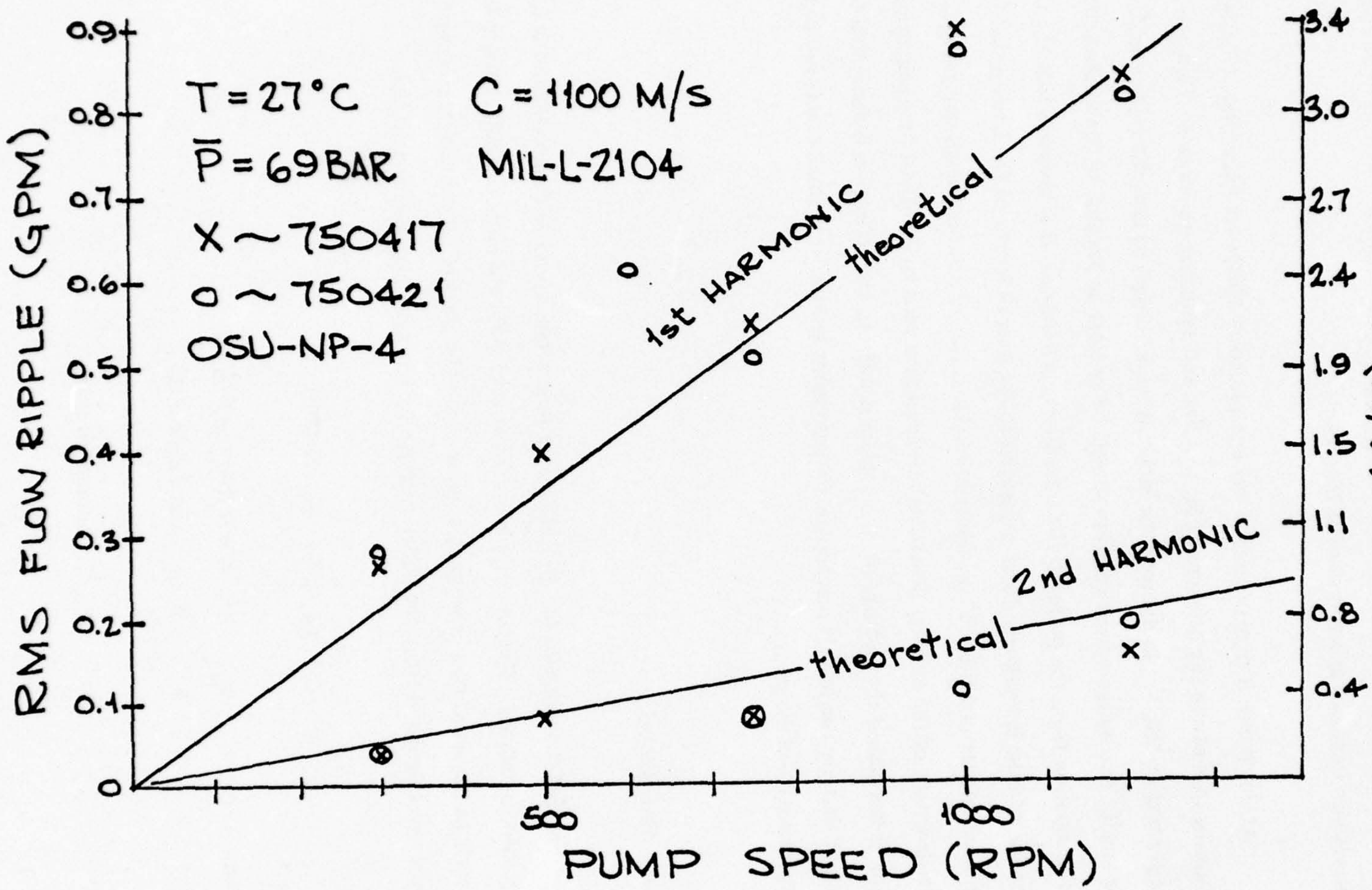


Fig. 2-3. OSU-NP-4 Flow Ripple Versus Pump Speed Showing Comparison Between Theoretical and Experimentally Derived Values.

R_g	=	base circle radius
$\dot{N}$	=	angular speed
w	=	$2\pi f$
f	=	frequency
t	=	time

The RMS value of the flow ripple at any particular harmonic will just be:

$$q_{m_{RMS}} = 0.707\, q_m \tag{2-10}$$

Theoretical flow ripples for OSU-NP-4 are shown in Table 2-1.

The experimental value of the flow ripple was obtained by measuring the pressure ripple and converting the pressure ripple to flow ripple using Eq. (2-1). Fig. 2-3 indicates that the fluidborne noise generated at the fundamental pumping frequency is significantly greater than that at the other harmonics. For instance, the first harmonic RMS flow ripple for OSU-NP-4 was, on the average, larger than the second harmonic by a factor of 5. Theoretically, the first harmonic should be 4.00 times larger than the second harmonic.

Pump Induced Pressure Measurements in a Non-Anechoic Environment

Appendix E contains data which were taken for a pump with different line impedances. The data in Appendix E vividly illustrate that the pressure ripple at a given distance from the pump outlet is significantly affected by the characteristics of the device loading the pump. Since theory [8] and experiment both indicate that fluidborne noise measurements are sensitive to the termination impedance, it should be clear that any test procedure must carefully control the system load impedance to insure adequate test code reproducibility.

TABLE 2-1. THEORETICAL FLOW RIPPLE FOR OSU-NP-4 FOR TEST CONDITIONS SHOWN IN FIG. 2-3.

N	HARMONIC	PREDICTED $\tilde{q}$ rms
800	1	0.2118
500	1	0.353
600	1	0.424
750	1	0.529
1000	1	0.706
1200	1	0.847
300	2	0.0529
500	2	0.0883
600	2	0.106
750	2	0.132
1000	2	0.176
12000	2	0.211

INDUSTRIAL INTERACTION

Project personnel have continued to interact with industry in the area of fluidborne noise. During the project year, personnel attended three national fluidborne noise standard meetings. The results of the pump-induced pressure measurements in non-anechoic environments were shared with the NFPA committee working on pump fluidborne noise generation potential (NFPA T3.9.24). In conjunction with project work with the NFPA, arrangements have been made with a prominent pump manufacturer to provide three pumps for fluidborne noise evaluation. These units will be used for *"round-robin"* evaluations of the procedure proposed by the committee.

SUMMARY

After considering test systems, instrumentation, and data analysis, it appears most practical to use an anechoic environment for obtaining experimental data to evaluate the

fluidborne noise generation potential of a hydraulic pump. Since all of the pumps evaluated at the Fluid Power Research Center have shown the trend of higher flow ripple at higher speeds, it is reasonable to expect that a number based on experimental results can be assigned to the coefficient a_m of Eq. (2-9). This means that it appears rational to use a single number rating at a specific outlet mean pressure to describe the flow ripple characteristics of a hydraulic pump. This single number would not reflect the output impedance of the pump.

The question of evaluating the impedance of the pump is being considered experimentally and will be discussed in the next quarterly report. The test procedure shown in Appendix A provides the basis for a practical means of evaluating the fluidborne noise generation potential of a hydraulic pump.

CHAPTER III

FLUIDBORNE NOISE ATTENUATOR EVALUATION

The acoustic power flow per unit of area in an infinitely long pipe is the rms pressure, $\tilde{p}$, times the rms particle velocity. The rms particle velocity is $\tilde{p}/Z$, where Z is the characteristic impedance of the pipe which is defined as $\rho C/S$, with S being the cross-sectional area of the pipe. The power in the pipe is [7]:

$$W = S\, p^2/\rho C \tag{3-1}$$

Infinitely long lines seldom exist in field hydraulic systems, so there is usually a right traveling wave, $(S\, p_i^2/\rho C)$, and a left traveling wave, $(S\, p_r^2/\rho C)$. The net power flow is:

$$W_{net} = (S/\rho C)\,(p_i^2 - p_r^2) \tag{3-2}$$

where p_i and p_r, respectively, are associated with the incident and reflected pressure waves. Eq. (3-2) can be shown to be equivalent to:

$$W_{net} = (S/\rho C)\,(p_{max})\,(p_{min}) \tag{3-3}$$

where p_{max} is the maximum pressure (rms) in the line and p_{min} is the minimum pressure in the line. The ratio p_{max}/p_{min} is defined as the standing wave ratio (SWR). The following equations show relationships between SWR, p_i, p_r, p_{max}, and p_{min}:

$$SWR = p_{max}/p_{min} = (p_i + p_r)\,/\,(p_i - p_r) \tag{3-4}$$

or rearranging:

$$p_r/p_i \quad = \quad (SWR - 1)/(SWR + 1) \quad = \quad (p_{max} - p_{min})/(p_{max} + p_{min}) \qquad (3\text{-}5)$$

Since $p_r = p_{max} - p_i$, Eq. (3-5) can be rewritten in terms of p_{max}, p_{min}, and p_i, yielding:

$$p_i \quad = \quad (p_{max} + p_{min})/2 \qquad (3\text{-}6)$$

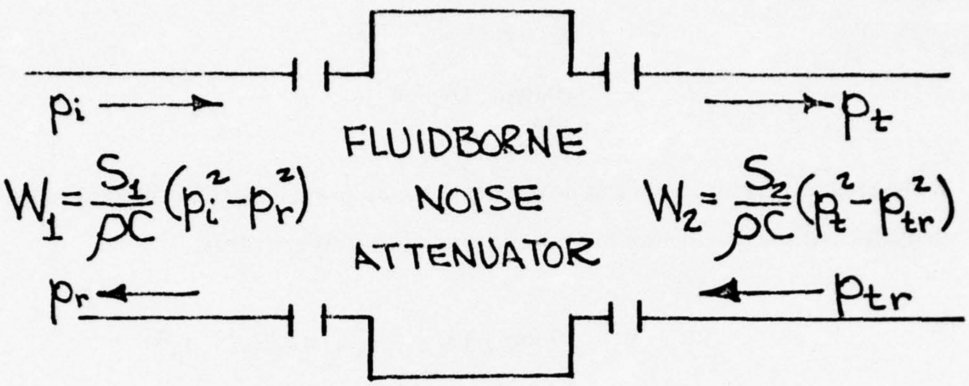
If $p_{min} << p_{max}$, then p_i is just $p_{max}/2$. When the reflection factor, R, is large, $p_{min}/p_{max} \approx 0$.

PROPOSED TEST PROCEDURE

A test code for the evaluation of fluidborne noise attenuators should consider the following characteristics: transmission loss, input impedance, output impedance, and flow resistance. The transmission loss indicates how much of the incident pressure ripple is emitted at the outlet of the attenuator. The input impedance indicates how much of the incident pressure is reflected by the attenuator toward the source. The output impedance indicates what percentage of the pressure incident on the attenuator outlet is reflected back to the downstream portion of the system. The flow resistance of the attenuator indicates the power dissipated by a given value of mean flow through the unit. These four characteristics adequately define the performance characteristics needed to acoustically rate the attenuator and to compare the power dissipated by the attenuator relative to the acoustical effectiveness of the unit.

Transmission Loss

Fig. 3-1 shows the relationship between p_i and p_r in a section of conduit connected to a fluidborne noise attenuator. Extending the same concept to the pipe downstream of the attenuator gives a relationship for the transmitted pressure, p_t, and the reflected pressure,



$$TL = 10 \log_{10} \frac{W_i}{W_t} \ (dB)$$

$$W_i = \frac{S_1}{\rho C} (p_i^{2})$$

$$W_t = \frac{S_2}{\rho C} (p_t^{2})$$

Fig. 3-1. Definition of Transmission Loss and Relationship Between Energy Flows to and from an FBN attenuator.

p_{tr}, in the downstream pipe. One performance parameter for the attenuator is transmission loss:

$$TL = 10 \log_{10} (W_i / W_t) \tag{3-7}$$

If the areas of the upstream and downstream conduits are equal and the system has equal densities and sonic velocities in both sections, then it follows that:

$$TL = 10 \log_{10} (S_1 p_i^2/\rho_1 C_1)(\rho_2 C_2/S_2 p_t^2) \tag{3-8}$$

$$TL = 10 \log_{10} (p_i^2 / p_t^2)$$

$$TL = 20 \log_{10} (p_i / p_t)$$

Thus, the transmission loss can be determined by evaluation of experimental data using Eqs. (3-6) and (3-8).

Input Impedance

Ignoring phase relationships for this report, we can consider only the real part of the input impedance of the attenuator that is the absolute value reflection coefficient, the reflection factor, R. The reflection factor is p_r/p_i. R can be evaluated using Eq. (3-5). The data for evaluating the input reflection factor is the same data used to evaluate the transmission loss.

Output Impedance

If the attenuator will accept flow through the unit from either direction, it is possible

to evaluate the output impedance using the output reflection factor, R_o. The data for calculating R_o is obtained by inverting the attenuator in the test circuit and obtaining the necessary measurements by repeating the test procedure used for obtaining the transmission loss and the input impedance.

If the pressure drop through the attenuator is less than or equal to an equivalent length of tubing, then the attenuator is essentially non-dissipative. For non-dissipative units, $p_t^2 = p_i^2 - p_r^2$, and Eq. (3-8) can be rewritten:

$$TL = 10 \log_{10} (p_i^2 / (p_i^2 - p_r^2)) \tag{3-9}$$

$$TL = 10 \log_{10} (1 / (1 - R^2))$$

Thus, the magnitude of the reflection factor for non-dissipative elements can be written in terms of the TL:

$$R^2 = 1 - 10^{-(TL/10)} \tag{3-10}$$

Pressure Drop

The efficiency of the attenuator can be evaluated by ratioing the attenuation of the unit to the pressure drop of the unit at the desired flow. To obtain an estimate of the efficiency of the attenuator,data are needed which show the pressure drop versus flow of the unit. These data can be obtained using any acceptable test procedure.

Frequency Range

One objective of using a fluidborne noise attenuator is to reduce the amplitude of system pressure ripple at the pumping frequency. Since most conventional pumps have

between 9 and 15 pumping elements (n) and are usually operated above 600 revolutions per minute (rpm) (N) but at or below 2500 rpm, the fundamental pumping frequency range is:

$$f_1 = Nn / 60 \qquad (3\text{-}11)$$

$$f_{low} = (600)\ 9/60 = 90 \text{ Hz.}$$

$$f_{high} = (2500)\ 15/60 = 625 \text{ Hz.}$$

The proposed procedure states that the first three pumping harmonics are to be measured, which will easily accommodate a frequency range of interest between 100 Hz. and 1000 Hz. This will allow testing to be conducted in the average fluid power laboratory, using maximum pump speeds in the vicinity of 2300 rpm for a pump with nine pumping elements, thus providing information about the attenuator over a frequency range which is important to the largest number of users.

Because the pressure level at the fundamental pumping frequency is usually more repeatable than the levels at higher harmonies and because most attenuators will probably respond best to the highest pressure level in the system, it is recommended that data for attenuator evaluation be taken at the first three pumping frequencies. Data at individual frequencies are obtained by varying the pump speed. At a minimum, the procedure requires measuring attenuator performance at the eleven third-octave frequencies between 100 and 1000 Hz.

TEST RESULTS

The attenuator selected for evaluation was an expansion chamber, which is a device with an increase in diameter at the inlet and a corresponding reduction in diameter at the outlet. The unit chosen for test was symmetrical, as described in Appendix F. The unit was tested

for attenuation characteristics, input impedance, and flow resistance. The results of the tests are described in the following paragraphs.

Anechoic Termination

The same anechoic termination was used for both the pump tests and the attenuator tests. The results of the evaluation of the anechoic termination are shown in Fig. 2-2. Fig. 3-2 shows the circuit used for evaluation of the attenuator.

Input Impedance

Fig. 3-3 shows the reflection factor for the input to the expansion chamber. The values for the reflection factor were obtained using the transmission loss and Eq. (3-10). Since the unit is symmetrical, tests were not conducted on the output impedance, which would be the same as the input impedance. The results of the pressure drop versus flow show that the assumption of the unit being non-dissipative is valid for the flow range examined.

Transmission Loss

Fig. 3-4 compares the results of the transmission loss evaluation with the theoretical transmission loss for the expansion chamber. The data scatter shown in Fig. 3-4 emphasize the need for averaging several evaluations at a given frequency to get a better estimate of the mean value of the transmission loss. The procedure recommended for evaluating the transmission loss requires taking three sets of data at each frequency and reporting the average value of the transmission loss. The transmission loss values shown in Fig. 3-4 are for first, second, and third harmonics of the fundamental pumping frequency.

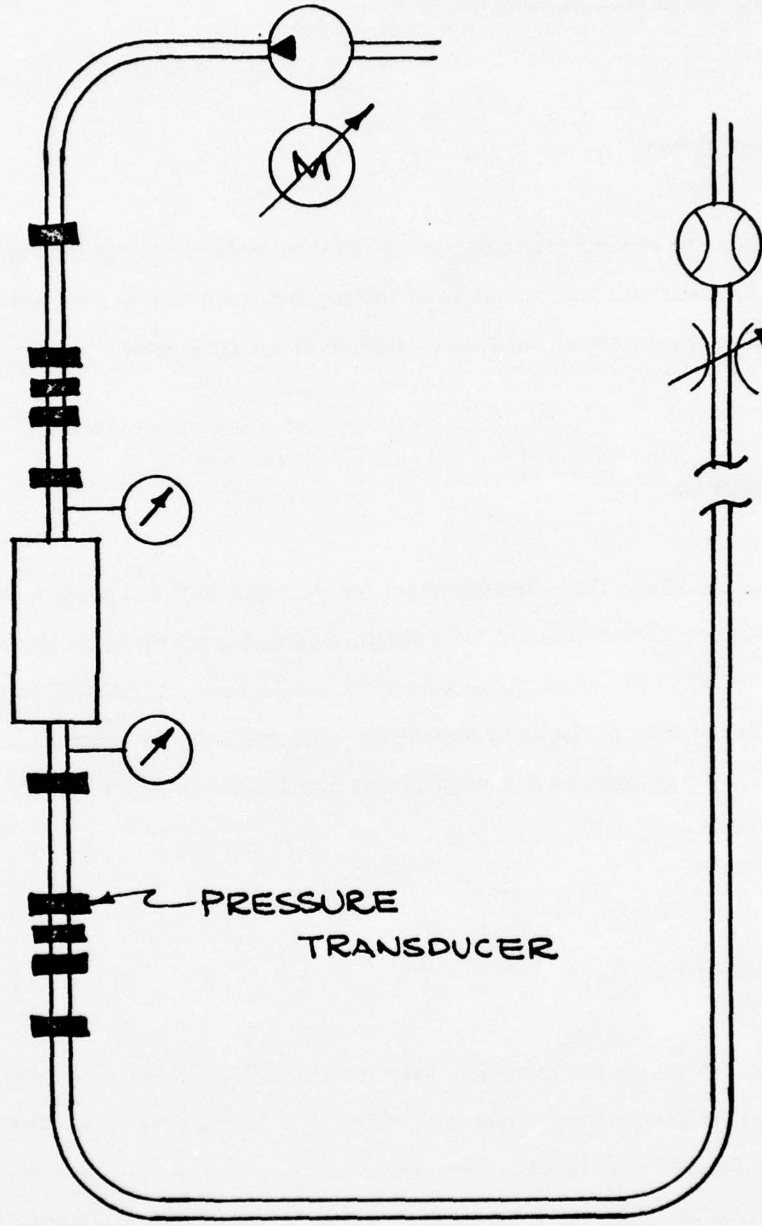


Fig. 3-2. Pressure-Ripple Attenuator Test System.

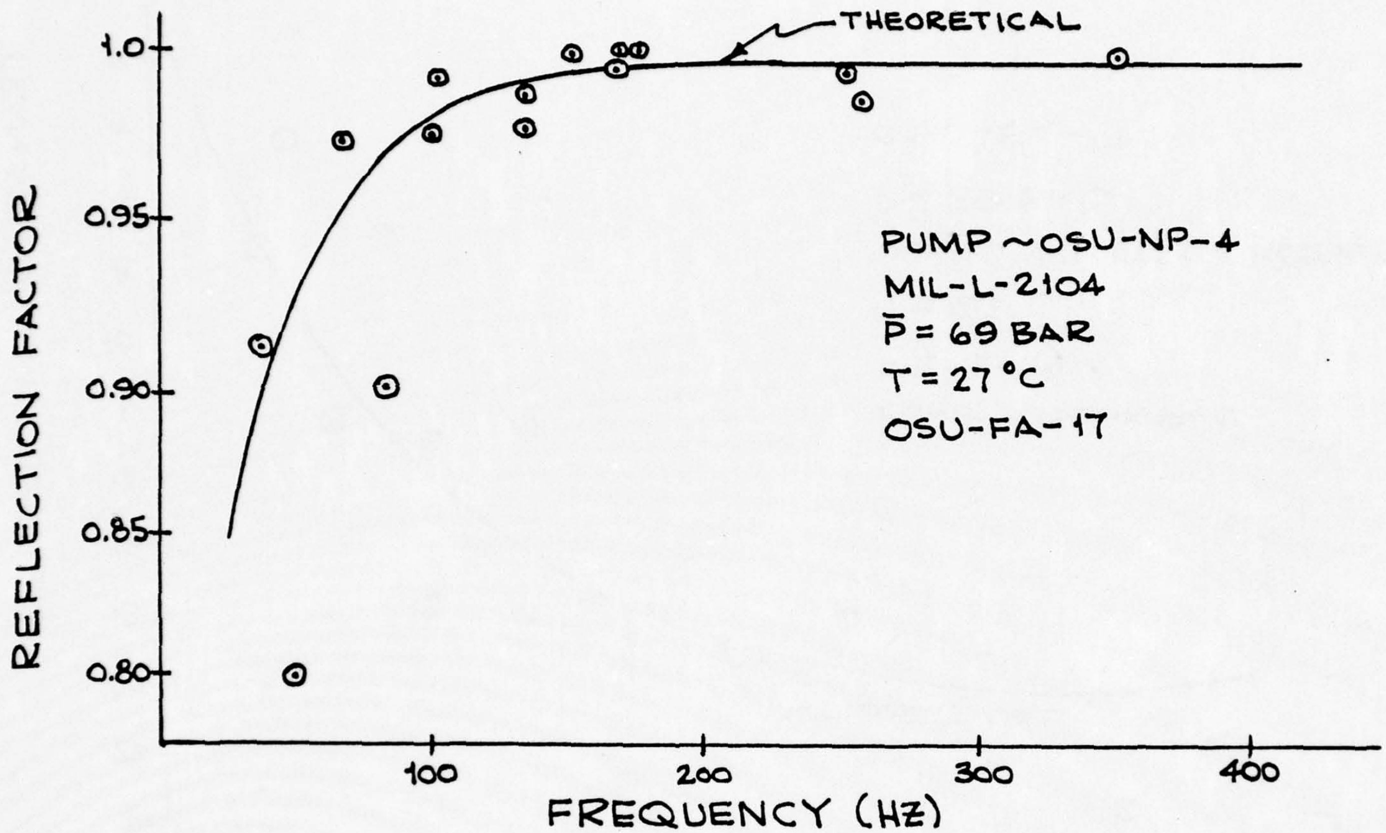


Fig. 3-3. Reflection Factor Versus Frequency Based on Transmission Loss. See Eq. (3-10).

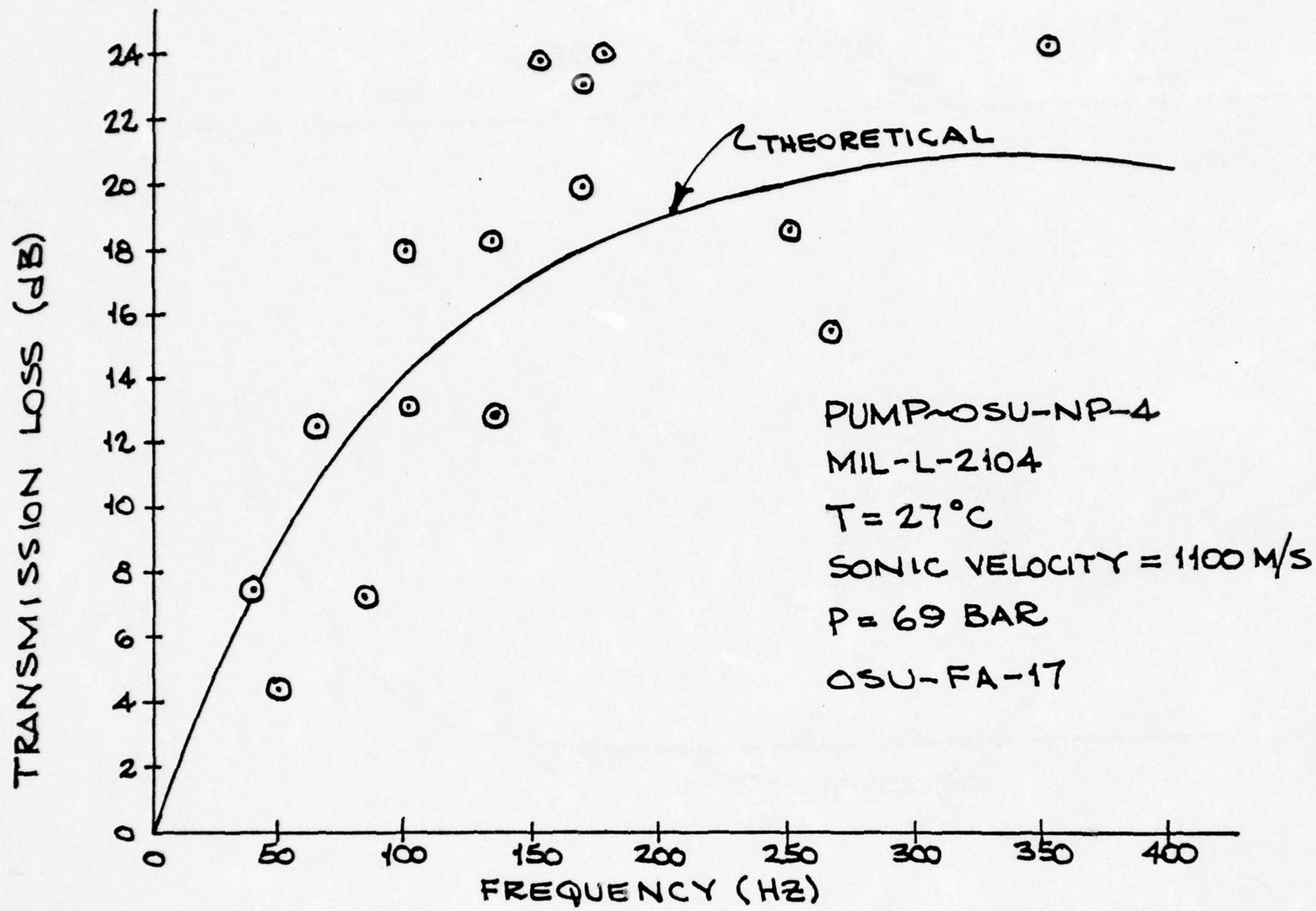


Fig. 3-4. Experimental Transmission Loss Based on Calculated Upstream Vs. Frequency Incident Pressure and Transmitted Pressure.

Pressure Drop

Fig. 3-5 illustrates the results of the flow resistance evaluation with the expansion chamber studied for this report. Since the pressure drop did not exceed 0.06 BAR, the assumption of the unit being non-dissipative for calculating the reflection factor using transmission loss is valid.

INDUSTRIAL INTERACTION

Industrial interaction in the area of fluidborne noise attenuators for this project year was limited to the presentation of a paper co-authored with Mr. S. E. Wehr of U.S. Army MERADCOM. The SAE paper, *"Fluidborne Noise Attenuator Performance Evaluation,"* [7] was well received. Feedback from interested members of industry provided guidance which has enhanced the procedure and made it more functional.

SUMMARY

The proposed test procedure for evaluating the performance of fluidborne noise attenuators has intentionally been written without a great amount of detail. The procedure, which is based on knowledge gained during this project year, needs to be validated. It appears that the general refinements will yield more reproducible results. Once it is shown that the standard deviation of the test results is reduced to an acceptable level, the procedure needs to be more explicitly outlined and presented to industry for evaluation. One area that needs to be carefully outlined is the computational procedure for reducing the raw experimental data to p_{max} and p_{min}.

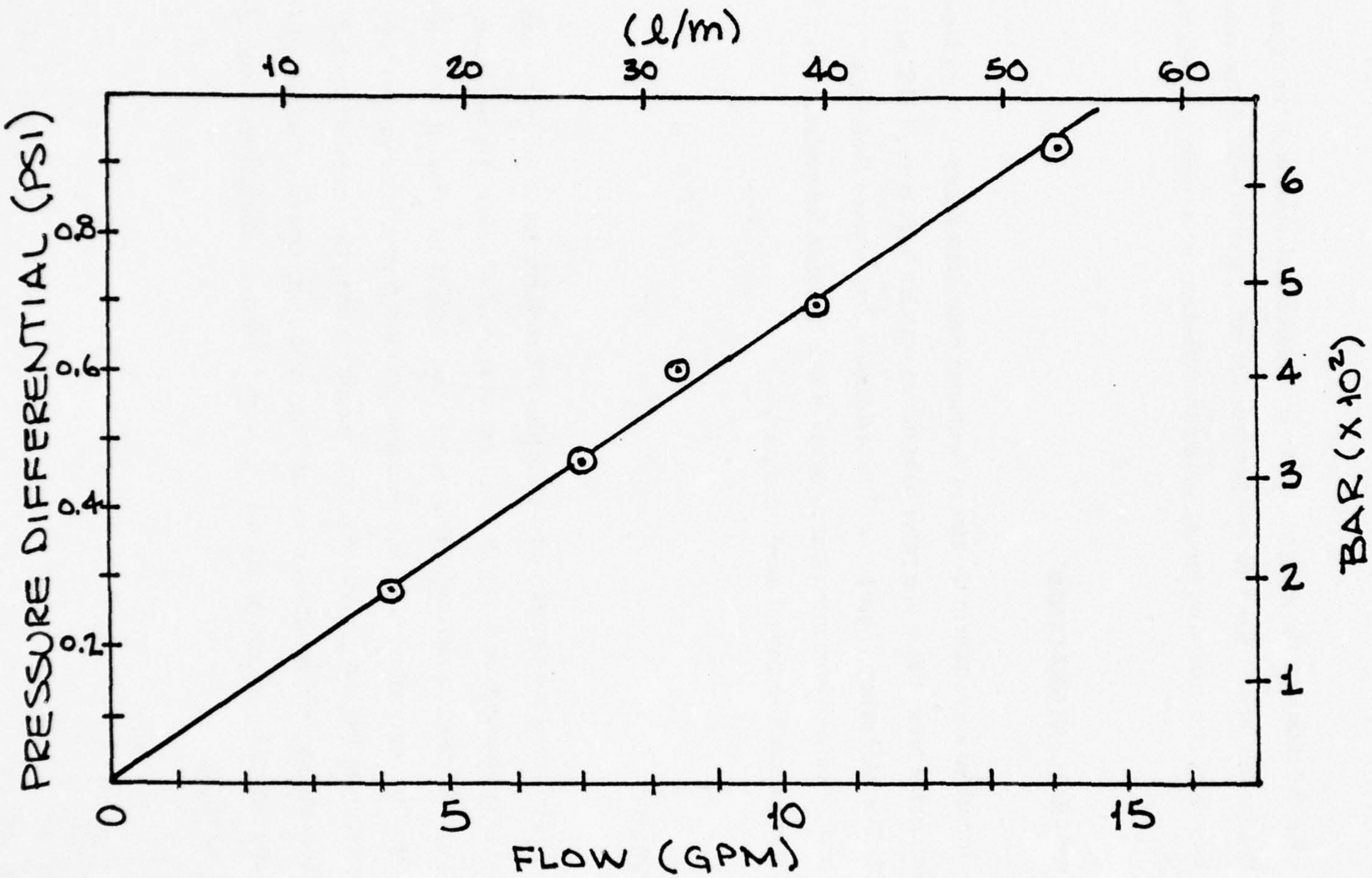


Fig. 3-5. Pressure Drop Versus Flow for Expansion Chamber OSU-FA-17.

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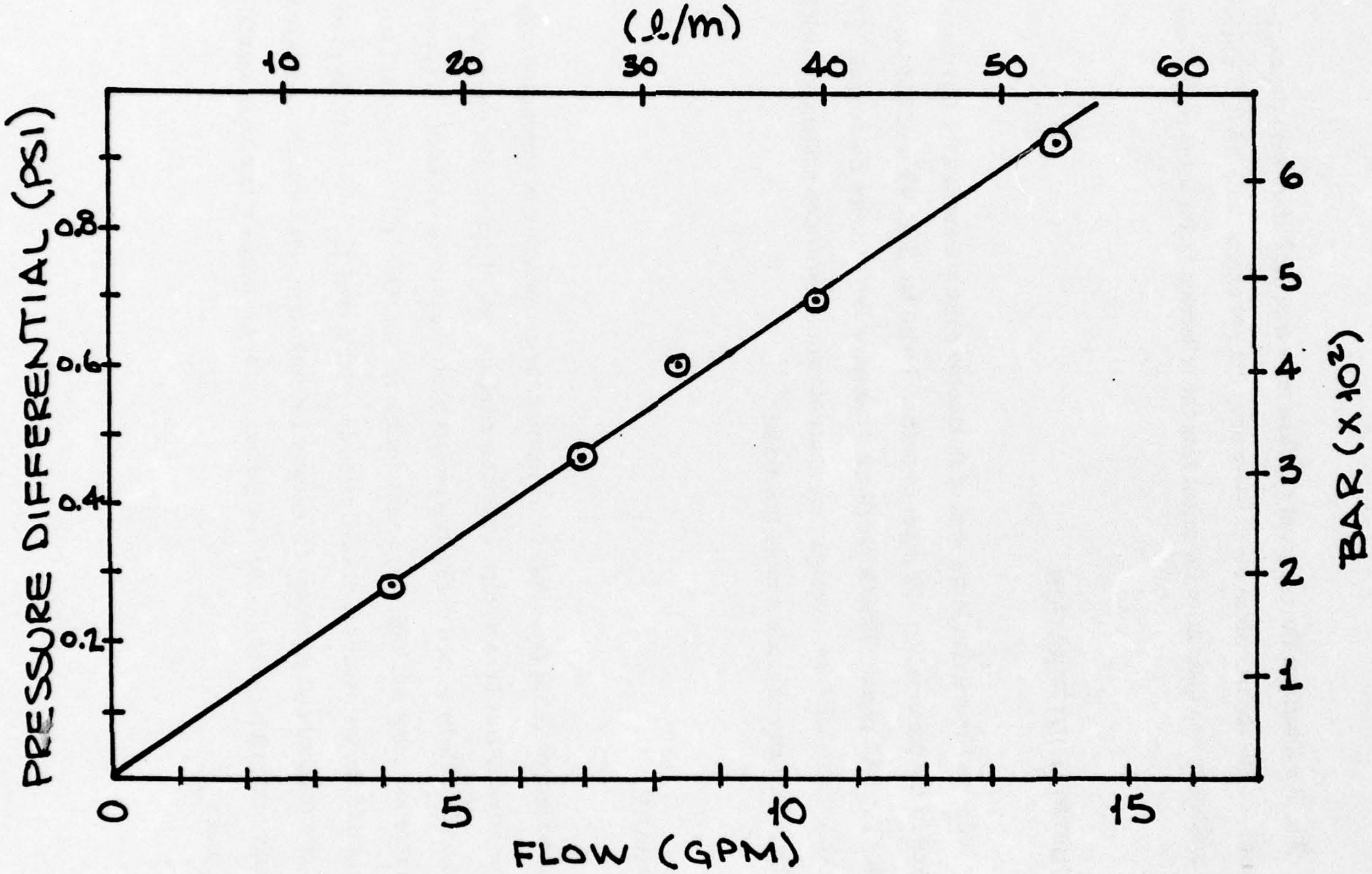

Fig. 3-5. Pressure Drop Versus Flow for Expansion Chamber OSU-FA-17.

CHAPTER IV

CONCLUSIONS AND RECOMMENDATIONS

Both of the proposed test procedures have been tested in the laboratory and modified to incorporate the results of the experimental validation as well as the inputs from industry. Both procedures form valid nuclei for industrially acceptable test codes.

The test code for evaluating the fluidborne noise generation potential of hydraulic pumps is based on the use of an anechoic termination in the hydraulic load system. This recommendation does not preclude the use of a procedure which requires the use of a non-anechoic termination in the test system. However, there are two important recommendations relative to the use of a non-anechoic termination for testing fluid power pumps. First, since data generated using the anechoic termination is much easier to handle mathematically and can easily lead directly to a specific characteristic of the tested unit, it seems more desirable to first develop a procedure using the anechoic termination, then work on the development of a procedure using non-anechoic terminations. Second, any procedure which uses a non-anechoic termination should have a validated conversion procedure to yield the same information that would be obtained using an anechoic termination; that is, the unique characteristic of the pump called the flow ripple.

For both test procedures, the mathematical manipulation of the experimental data could be programmed on a computer or at least clearly outlined to allow personnel unfamiliar with the computation procedures to obtain the desired performance characteristics.

For all fluidborne noise measurements and predictions in hydraulic systems, a note of caution should be observed regarding the magnitude of the oscillatory pressures, which are

usually significant relative to the mean pressure levels. Nonlinearities resulting because of the high noise/mean pressure ratios will become more evident as we gain more knowledge in this area. They may not be a major concern.

Both of the proposed procedures should be pursued to completion. Both procedures are only a few steps away from a complete package that can be critiqued by industry.

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APPENDIX A

PUMP FLUIDBORNE NOISE

TEST PROCEDURE

APPENDIX A

PUMP FLUIDBORNE NOISE TEST PROCEDURE

1. Mount pump and pressure transducers with anechoic termination. (See Fig. 2-1.)

2. Test anechoic termination:

   a. Set mean system pressure to maximum rating.

   b. Insure that the range of dB readings $\leq$ 4 dB at each frequency over the desired frequency range.

3. Measurement of pump flow ripple:

   a. After establishing condition 2(b), test the pump at the following speeds and maximum pressure: one-fourth, one-half, three-fourths, and maximum speed. Record the first three harmonics for each setting.

   b. Repeat 3(a) for one-half maximum pressure.

4. Measurement of pump impedance:

   a. Place a known impedance load downstream of the pressure transducers.

   b. Measure the distances between the pump and the load and the pump and the transducers.

   c. Repeat 3(a) and 3(b) for the given speeds and pressure conditions.

5. Reducing and reporting the data:

   a. From the tests in Part 2, determine the SWR and reflection factor for the anechoic termination. The data should be plotted in the same format as Fig. 2-2.

   b. From the tests in Part 3, determine the value of the pump flow ripple, $\tilde{q}$, using Eq. (2-1). The flow ripple should be plotted as a function of the speed of the pump with three different harmonics. (See Fig. 3-1.)

The overall value of the flow ripple can be obtained for each speed by adding q_1, q_2, and q_3 at each speed. A value of q/N should be obtained by performing a linear regression on the plot of q_{TOT} vs. speed.

c. From the tests in Part 4, determine the impedance of the pump as a function of frequency at different pressures.

TABLE A-1

DATA RECORDING TABLE FOR PUMP FLOW RIPPLE TEST

DATE ________ FLUID TEMP ________ SONIC VELOCITY ________

PUMP I.D. ________ FLUID ________ MAXIMUM OUTLET PRESSURE ________

MAXIMUM PUMP SPEED ________ FLUID VISCOSITY ________

PUMP FLOW RIPPLE (ℓ/m)						
SPEED	PRESSURE				FREQUENCY	
	0.25	0.50	0.75	1.00		
0.25					f_1	
					f_2	
					f_3	
0.50					f_1	
					f_2	
					f_3	
0.75					f_1	
					f_2	
					f_3	
1.00					f_1	
					f_2	
					f_3	

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APPENDIX B

FBN ATTENUATOR

TEST PROCEDURE

APPENDIX B

FBN ATTENUATOR TEST PROCEDURE

Fig. 3-2 depicts the test system for the test procedure, which consists of the following basic steps:

1. Without the attenuator installed:
   a. Insure that the range of dB readings $\leq$ 4 dB.
   b. Measure a *"tare"* pressure drop versus flow.

2. With pressure transducers installed upstream of the attenuator, conduct the following tests for each pumping frequency shown in Table B-1:
   a. Record data for the first three harmonics of the pumping frequency for a constant system pressure.
   b. Measure pressure drop across attenuator and flow.
   c. Slowly relieve the system pressure to a minimum pressure.
   d. Increase system pressure to that in 2(a).
   e. Repeat measurements in 2(a).
   f. Slowly relieve the system pressure to a minimum pressure.
   g. Increase system pressure to that in 2(a).
   h. Repeat measurements in 2(a).

3. Install pressure transducers downstream of the attenuator.
   a. Repeat measurements in 2(a) and 2(c) through 2(h).
      NOTE: If sufficient transducers are available, these measurements may be taken simultaneously with 2(a), 2(c)-2(h).

4. Reduce the data.
   a. Obtain pressure drop versus flow due to the attenuator by correcting 2(b) with 1(b).

5. Obtain mean values for the experimental data and report the reduced data in a form similar to that shown in Figs. 3-3, 3-4, and 3-5.

TABLE B-1. TABLE OF TEST FREQUENCIES USING TEST PLAN FOR OBTAINING 15 DATA POINTS FOR EVALUATING ATTENUATOR PERFORMANCE.

Desired 1/3 Octave Center Frequency (Hz)	TEST NUMBER				
	1	2	3	4	5
100	100*				
125		125*			
160			160*		
200	200				
250		250		(267)*	
315	(300)		(320)		315*
400		(375)			
500			(480)	(533)	
630					630
800				800	
1000					(945)

*Fundamental puming frequencies.

NOTE: Frequencies in parentheses deviate from desired frequencies.

APPENDIX C

ACOUSTICAL DATA REDUCTION

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APPENDIX C

ACOUSTICAL DATA REDUCTION

The data reduction programs included in this appendix were used to facilitate converting fluidborne noise data in dB to pressures. This appendix contains the following programs:

C-1	Recorder dB to PSI, then N/M^2 (HP-25)
C-2	Recorder dB to PSI, then BAR (HP-25)

TABLE C-1. HP-25 PROGRAM TO CONVERT RECORDER dB TO PSI, THEN N/M².

Data Input: dB or Chart			
ENTER	01		31
4	02		04
2	03		02
.	04		73
7	05		07
–	06		41
2	07		02
0	08		00
÷	09		71
1	10		01
0	11		00
x y	12		21
y^x	13	14	03
2	14		02
.	15		73
9	16		09
x	17		61
Pause	18	14	74
Pause	19	14	74
Pause	20	14	74
1	21		01
.	22		73
4	23		04
5	24		05
EE	25		33
4	26		04
CHS	27		32
÷	28		71
GTO 00	29	13	00

DATA OUTPUT: PSI IS DISPLAYED FIRST, THEN THE N/M² EQUIVALENT OF THE RECORDED dB.

1 N/M² = 1 / 1.45 x 10 PSI

NOTE: Recorder calibrated per FBN calibration procedure in Appendix G.

TABLE C-2. RECORDER dB TO PSI, THEN BAR.

DATA INPUT: dB ON CHART

	DISPLAY		
KEY ENTRY	**LINE**		**CODE**
ENTER	01		31
4	02		04
2	03		02
·	04		73
7	05		07
−	06		41
2	07		02
0	08		00
÷	09		71
1	10		01
0	11		00
x y	12		21
f y^x	13	14	03
2	14		02
·	15		73
9	16		09
x	17		61
f Pause	18	14	74
f Pause	19	14	74
f Pause	20	14	74
1	21		01
4	22		04
·	23		73
5	24		05
÷	25		71
GTO 00	26	13	00

DATA OUTPUT: PSI IS DISPLAYED FIRST, THEN THE BAR EQUIVALENT OF THE CHART dB.

NOTE: Recorder calibrated per FBN calibration procedure in Appendix G.

APPENDIX D

INSTRUMENTATION

APPENDIX D

INSTRUMENTATION

I. GENERAL RADIO

A.	1523	Level Recorder
B.	1523-P1	Preamplifier Plug In
C.	1523-P3	1/3 Octave Band Analyzer
D.	1523-P4	Wave Analyzer
E.	1523-9622	50 dB Potentiometer
F.	1560-9531	Microphone
G.	1560-9580	Tripod
H.	1560-9666	Microphone Cable
I.	1560-P42	Microphone Preamplifier
J.	1562-A	Sound Level Calibrator
K.	1933	Precision Sound Level Meter and Octave Band Analyzer
L.	130BR	Oscilloscope

II. HEWLETT-PACKARD

A.	HP-25	Programmable Calculator
B.	HP-55	Programmable Calculator

III. TEKTRONIX

A. 502	....	Dual-Beam Oscilloscope
B. RM31A	....	Oscilloscope

IV. PCB PIEZOTRONICS, INC.

A. 111A24	....	Quartz Pressure Transducer
1. SN 670	....	5.14 mv/psi
2. SN 671	....	5.10 mv/psi
3. SN 672	....	5.18 mv/psi
4. SN 673	....	5.32 mv/psi
5. SN 674	....	5.30 mv/psi
6. SN 675	....	5.35 mv/psi
7. SN 676	....	5.20 mv/psi
B. 483M20	....	ICP Power Supply

V. TEAC

A. 1230	....	Tape Deck

VI. BECKMAN

A. 7370R	....	Universal EPUT Meter

APPENDIX E

FLUIDBORNE NOISE MEASUREMENTS IN A NON-ANECHOIC ENVIRONMENT

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APPENDIX E

FLUIDBORNE NOISE MEASUREMENTS IN A NON-ANECHOIC ENVIRONMENT

The effect of varying the impedance downstream of a hydraulic pump on the pressure at a given distance from the pump is illustrated by the results of a series of tests conducted in conjunction with this project. Fig. E-1 shows schematically the different system load configurations used for the tests. Fig. E-2 summarizes the data from the measurement tests. Table E-1 lists the average differences between the reference test system as well as the maximum difference between the tests. The results of these tests clearly illustrate the importance of carefully controlling the system impedance or carefully accounting for impedance differences between laboratories if it is desired to develop a test procedure that will insure acceptable reproducibility.

TABLE E-1. MAXIMUM AND AVERAGE PRESSURE LEVEL DIFFERENCES FOR EACH MEASUREMENT CASE.

CASE	AVERAGE DIFFERENCE (dB)	MAXIMUM DIFFERENCE (dB)
I	5.2	10.0
IIA (Reference)	0.0	0.0
IIB	0.2	1.0
III	2.5	5.5
IV	0.7	2.0

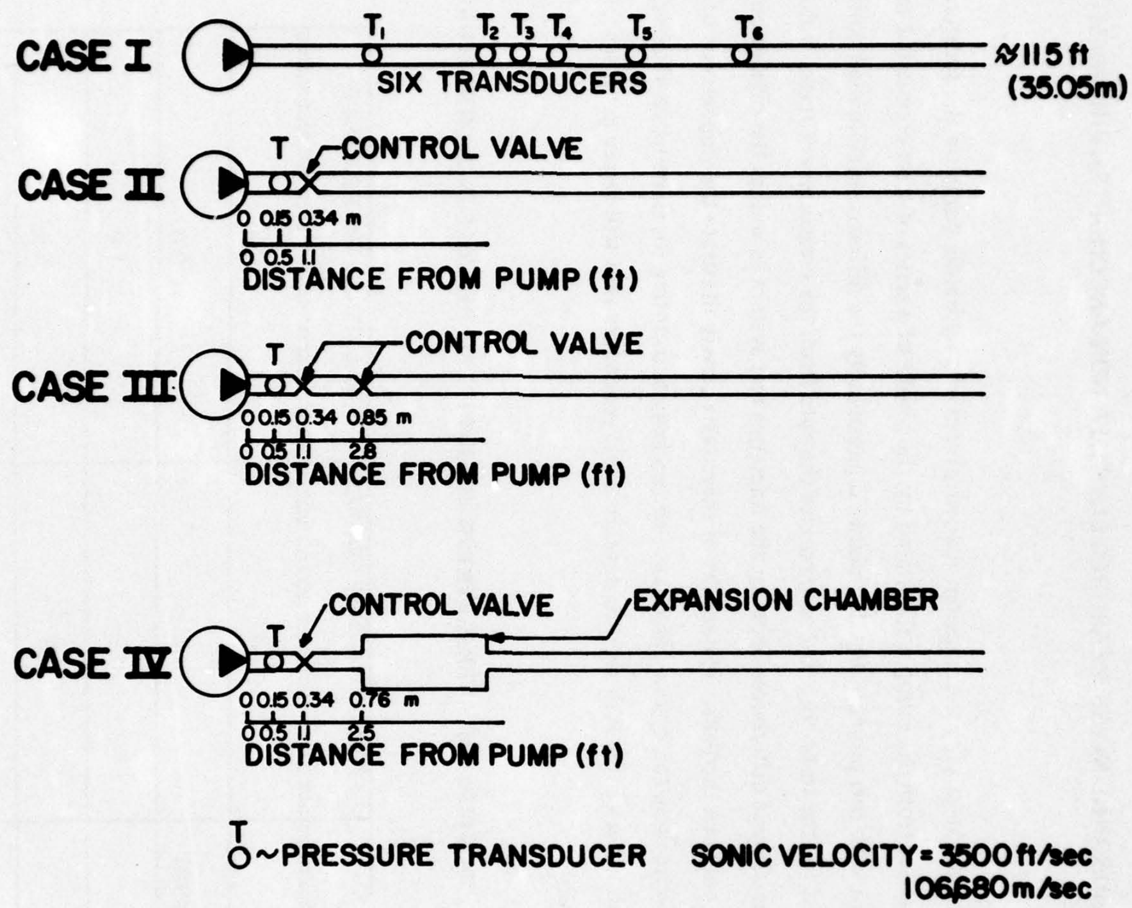

Fig. E-1. Four Test Systems Used to Examine the Effect of System Configuration on "Near-Field" Pump Outlet Pressure Ripple.

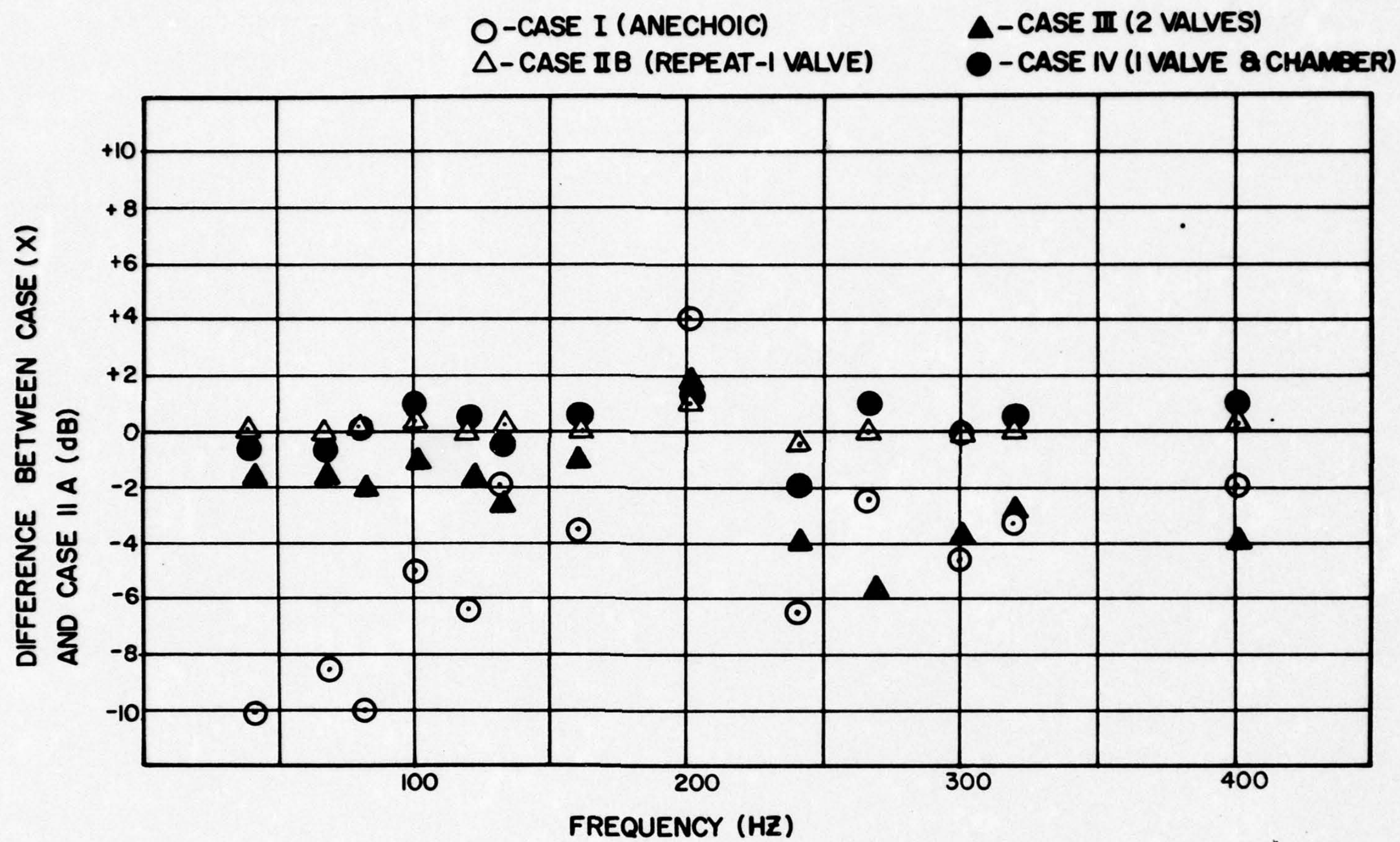

Fig. E-2. Plot of Data Showing Effect of Different System Configurations on "Near-Field" Pump Outlet Pressure Measurements.

APPENDIX F

CRITICAL TEST COMPONENTS

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APPENDIX F

CRITICAL TEST COMPONENTS

The principal components used for the tests reported in this section were an external gear pump, an *"expansion -chamber"* attenuator, and an anechoic termination. These components have the following basic characteristics:

GEAR PUMP

Base Circle Radius = 19.8mm (0.7792 in.)

Pitch Circle Radius = 22.2mm (0.8745 in.)

Addendum Circle Radius = 26.9mm (1.0605 in.)

Face Width of Gear = 38.5mm (1.512 in.)

Number of Teeth = 10

EXPANSION CHAMBER ATTENUATOR

Length = 914mm (36.0 in.)

Inside Diameter = 103mm (4.05 in.)

Inlet Diameter = 21.6mm (0.85 in.)

ANECHOIC TERMINATION

Configuration: 20.7M (68 ft.) of 25.4mm (1 in.) steel tubing + 31.1M (102 ft.) of 25.4mm (1 in.) hose (Titeflex, 400S5247-1440) + 4.0M (13 ft.) of 25.4mm (1 in.) steel tubing

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APPENDIX G

CALIBRATION

APPENDIX G

CALIBRATION

In this appendix, a step-by-step procedure is outlined for calibrating the analysis instrumentation and then converting dB to psi when taking fluidborne noise measurements.

Three techniques are discussed:

1. **Reference Voltage Technique – A reference voltage source with a known output is used for calibration.**
2. **Microphone Technique – A known, calibrated sound pressure level source is attached to a microphone and the resultant voltage is used for calibration. An analyzer is used as a recorder.**
3. **Analyzer Reference Technique – The analyzer reference voltage is used as a reference for an unknown voltage source.**

Technique 1 is discussed in Section A of this appendix. Both 2 and 3 use a constant voltage source (for 2, the voltage from the microphone will correspond to the sound pressure level from the reference source). Those two methods are similar and are discussed together in Section B.

SECTION A – REFERENCE VOLTAGE TECHNIQUE

Step I – Determining the Base Voltage in the Analyzer

Since the reference voltage is known (for example, V_{REF} = 1.061 volts for the GR 1562-A), the base voltage of the analyzer can be determined. The base voltage is first adjusted so that the GR 1523 analyzer reads 40 dB with 40 dB attenuation. The base voltage is then calculated:

$$(40 + 40) = 20 \log_{10} \frac{1.061}{V_B} \qquad V_B = 1.061 \times 10^{-4} \text{ volts}$$

Step II – Relationship Between dB and PSI

With the base voltage known and the voltage gain given for the pressure transducers, the relationship between PSI and dB can be obtained.

First, if the gain of the pressure measurement device is .005V/PSI, then for every PSI there is .005 volts:

<u>ON CHART</u>	<u>IN FLUID</u>
.005V	1 PSI
$dB = 20 \log_{10} \frac{.005V}{.0001061V}$	$dB = 20 \log_{10} \frac{1 \text{ (PSI)}}{2.9 \times 10^{-9} \text{ (PSI)}^*}$
$dB = 33.47$	$dB = 170.8$

*2.9×10^{-9} PSI equals $20 \mu N/M^2$

So, 33.47 dB on the recorded chart is equivalent to 170.8 dB in the fluid. Or, if a reading is taken on the chart, 138.3 must be added to the reading to get the actual pressure level in the fluid.

SECTION B – MICROPHONE AND ANALYZER REFERENCE TECHNIQUE

Step I – Determining Calibrator Voltage

With the GR 1523 analyzer in the *"calibrate"* position, the base voltage, V_b, in the analyzer is $100\mu V$. A constant voltage is used, which when connected to the analyzer yields a recorder level consistent with the input voltage; for example, for a GR 1562-A, approximately 41.5 on the chart (with 40 dB attenuation). So, the input voltage can be determined. For example:

$$(40 + 41.5) = 20 \log_{10} \frac{V_{CAL}}{V_B}$$

$$V_{CAL} = 10^{\frac{81.5}{20}} (100\mu V)$$

$$V_{CAL} = 1.188 \text{ volts}$$

Step II – Base Voltage

Now, the base voltage is adjusted to a convenient level, for the example, so that the analyzer reads 40 instead of 41.5. The new base voltage can then be calculated. For the example:

$$(40 + 40) = 20 \log_{10} \frac{1.188}{V_B}$$

$$V_B = 1.188 \times 10^{-4} \text{ volts}$$

Step III – Relationship Between dB and PSI

With the base voltage known and the voltage gain given for the pressure measurement system, the relationship between PSI and dB can be obtained. First, it is known that the gain of the system is .005V/PSI. So, for every PSI, there is .005 volts:

ON CHART | **IN FLUID**

.005V | 1 PSI

$$dB = 20 \log_{10} \frac{.005V}{.0001188} \qquad dB = 20 \log \frac{1}{2.9 \times 10^{-9}}$$

$$dB = 32.48 \qquad dB = 170.8$$

So, 32.48 dB on the chart is equivalent to 170.8 dB in the fluid. Or, if a reading is taken on the chart, 138.3 dB must be added to the reading to get the actual pressure level in the fluid. The same procedure for the microphone is shown in Table G-1.

To convert a pressure level in dB to PSI, take the reading, N, off the chart, add 138.3 dB to it, then use the following to get the RMS PSI:

$$(N + 138.3)\ dB = 20 \log_{10} \frac{P_{RMS}}{2.9 \times 10^{-9}}$$

If the pressure is sinusoidal, the peak-to-peak pressure amplitude is:

$$P_{p\text{-}p} = 2.83\ P_{RMS}$$

TABLE G-1. FBN MEASUREMENT CORRECTION (Using ABN Instrumentation Calibration).

GIVEN:	MICROPHONE (if used)	½" GR #612	______
	MICROPHONE CALIBRATOR LEVEL, L_c	114dB	______
	FBN INSTRUMENTATION GAIN --, G_f	4.7 mV/psi	______
	INSTRUMENT REFERENCE LEVEL, V_i	100 μV	______

EXAMPLE	ACTUAL
1. Output Voltage, V_c, wl/or with calibrator "ON" and instrument set to V_i	
Instrument reading, L_1 84.8 dB	Instrument reading, L_1 ______ dB
$(L_1) = 20 \log_{10} \frac{V_c}{(V_i)}$	$(\quad) = 20 \log_{10} \frac{V_c}{(\quad)}$
V_c = 1.737 v	V_c = ______
2. Instrument reference voltage, V_r, after calibration	
Instrument reading during calibration, L_c 84 dB	Instrument reading during calibration, L_c ______ dB
$(L_c) = 20 \log_{10} \frac{(V_c)}{V_r}$	$(\quad) = 20 \log_{10} \frac{(\quad)}{V_r}$
V_r = 109 μv	V_r = ______
3. Instrument reading, L_2, with FBN input of 1 psi	
$L_2 = 20 \log_{10} \frac{(G_f)}{(V_r)}$	$L_2 = 20 \log_{10} \frac{(\quad)}{(\quad)}$
L_2 = 32.7 dB	L_2 = ______
4. Correction to reference FBN to 20 μN/M²	
1 psi r_e 20 μPa 170.8 dB	(1 psi r_e 20 N/M²) 170.8 dB
Subtract L_2 − 32.7 dB	(Subtract L_2) − ______
CORRECTION 138.1 dB	CORRECTION * ______
	*Add to instrument reading to obtain FBN r_e 20 μPa

SECTION III

HYDRAULIC SYSTEM DIAGNOSTICS

PROJECT STAFF

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FOREWORD

This section presents the results, interpretation, and conclusions acquired by the effort associated with the area of system diagnostics. Specifically, the activity was directed toward the investigation and adaptation of Ferrographic fluid analysis for use with hydraulic systems. Laboratory tests were conducted to gain an appraisal of the capabilities and limitations of Ferrographic analysis and to determine the characteristic patterns of the wear debris. The data from these tests are typified in the report, while a complete summary is included in the appendices.

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LIST OF FIGURES

CHAPTER I

INTRODUCTION

When a machine operator starts his vehicle, he fully expects it to perform the functions for which it was designed in an effective manner. However, due to the wear of critical machine parts, the performance of the machine will slowly degrade. In some cases, the period of slow deterioration is concluded when one or more of the parts breaks and totally incapacitates the machine. Thus, there are two seemly distinct types of machine failures (slow but steady performance degradation and breakage) but which, in fact, have a common cause – surface deterioration.

Wear in systems where fluid is circulated is one of the most costly aspects involved in their operation. Many millions of dollars are spent yearly for maintenance in an attempt to prevent wear or to replace parts and components which have worn out or broken as a result of wear. In the past ten years, there has been an ever-increasing recognition that a significant portion of the wear in such fluid systems can be traced directly to the presence of particulate contaminant.

Contaminant wear in fluid systems, particularly hydraulic systems, is impossible to study directly. Therefore, investigators in the past have either evaluated the end results of the wearing process (dimensional changes or weight loss) or have measured the amount of performance degradation due to a particular contaminant environment. This latter approach has proven to be the most successful, and standard test procedures have been developed to evaluate the contaminant sensitivity of various hydraulic components based upon performance changes.

There are several drawbacks, however, to this type of contaminant wear assessment. First of all, the most obvious disadvantage is that the component is totally destroyed during the test.

Second, the technique does not provide any diagnostic information. That is, although based upon the results of a contaminant sensitivity test it is possible to estimate component life under certain contaminant exposures, there is no way to diagnose the internal state of a system. In other words, the missing diagnostic link is a technique whereby early warning signs could be used to prevent untimely field failures.

Numerous attempts have been made to utilize spectrographic techniques to provide the necessary diagnostic information associated with fluid systems. This method has apparently worked satisfactorily for lubrication systems but left something to be desired for hydraulic systems. Therefore, when a new technology was introduced in 1974 which seemed to hold promise of usefulness in diagnosing hydraulic systems, the Fluid Power Research Center initiated this program to fully develop the area [1, 2, 3]. Called Ferrography, this new technology basically utilizes an appraisal of the wear debris generated by the wear process to provide the desired diagnostic information.

The activity reported herein was designed to develop the Ferrographic technique into a useful engineering tool and was organized under a Technology Development Project (TDP). This type of project is used for efforts where interest is specialized within the industry, no pure research *per se* is required, or the necessary funding is beyond the scope of other project activities. Technology Development Projects are designed to satisfy specific objectives and are characterized by the following features:

1. Activity is not of a pure research nature, since seeding by previous investigations has demonstrated its feasibility (not blue sky).
2. Knowledgeable industrial advisors are already available for performing the necessary liaison to insure the success.
3. Direction and extent of project can be fully described through the assistance of the advisors.
4. Project objectives require more extensive budgetary and manpower commitments than can be allocated from the BFPR Program.
5. Project must appeal to enough companies to make it economically feasible.

6. Competent technical personnel are available to conduct the project.

7. Results of the project will yield engineering application tools having economic significance to the sponsors.

A unique feature of the Technology Development Project is that it is normally a joint activity. In the case of Hydraulic System Diagnostics, there are five sponsoring groups (four industrial and MERDC). The four industrial sponsors are John Deere, Massey-Ferguson, J. I. Case, and International Harvester. These sponsors have provided the necessary guidance, test components, and incentive for the project activity. Since this particular TDP included the U.S. Army MERDC on a matching funds basis, the reports which are generated throughout the effort will be subject to public disclosure.

This section of the report describes the work that has been accomplished on this diagnostic project. The test results have been analyzed in both an objective and subjective manner. The conclusions which have been reached at this time are presented and the plans for future efforts delineated. The results obtained to date are certainly encouraging, and it is anticipated that the project will be continued.

CHAPTER II

SCOPE OF EFFORT

As proposed, the prime intent of this Technology Development Project is to develop a method for applying the Ferrographic technology to the evaluation of hydraulic systems. The proposed plan of attack called for conducting laboratory tests on a number of hydraulic components and performing Ferrographic evaluation of the test results. From this Ferrographic analysis, an atlas would be compiled which would pictorially illustrate the wear debris patterns associated with each type of component exposure.

Once sufficient information has been accumulated from the laboratory tests to identify the various wear modes and their severity, the project would move into a field system evaluation phase. Several systems would be selected with the guidance of the project sponsors. These systems would be comprised of components for which sufficient wear information is available. The systems would be sampled periodically and appraised Ferrographically to acquire the necessary field system data. In addition, the operational conditions to which the system is exposed during field services would be monitored. The Ferrographic analysis of the field system would be compared with the information contained in the Ferrographic atlas in order to diagnose the internal state of the selected system.

At this point in time, the project activities have been concentrated upon the analysis of wear in hydraulic pumps. In all, 12 pumps have been submitted. These tests included break-in assessment (where no contaminant was added) as well as the normal 300 mg/litre contaminant sensitivity tests [4, 5]. This report contains all of the information collected from these pump tests. It is anticipated that additional component tests will be included in the continuation of this project.

The logic behind the approach taken in this project lies in the desire to develop the Ferrographic system into a worthwhile engineering tool. It stands to reason that, if a technique can be successfully utilized to assess the internal state of a hydraulic component or system operating under field conditions, it could also be used as a laboratory test monitor. Also, in order to insure success of any field system diagnostic attempt, it is necessary to investigate the wear debris patterns which are produced from a hydraulic component under various environments. Thus, the laboratory test phase was initiated first to become familiar with the Ferrographic system and to determine its characteristics and its limitations. Pumps were selected as the first test component because of their reputation as the most critical component in a hydraulic system.

The next section of this report is concerned with the thorough description of the Ferrographic system together with a brief review of other techniques which have been used for diagnostic purposes. The remainder of the report contains test results, analysis of results, conclusions, and recommendations. The data summary sheets containing all the test data are included in the appendices of this section.

CHAPTER III

WEAR ANALYSIS AND DIAGNOSTICS

Without a doubt, the cost factor has been responsible for the emergence of fluid power diagnostic techniques in recent years. As systems have grown in complexity, the cost of their maintenance and down time as well as their initial cost have become very large. It is no longer practical to assign a maintenance engineer to periodically dismantle the system and inspect the various component surfaces for evidence of wear. However, the fact remains that, in order to perform a satisfactory diagnosis of a hydraulic system, it is necessary to evaluate the severity of the wear processes active in the system. Furthermore, to offer a remedy for a severe wear condition, it is necessary to have some knowledge concerning the dominant wear mode and the possible location of the wearing surface or surfaces.

In a hydraulic system, it is impossible to measure the wear process directly. Therefore, engineers have been forced to evaluate wear conditions indirectly. Until recently, the most popular wear indicator and therefore diagnostic tool has been the spectrograph. In general, there are two types of these instruments – the emission type [3] and the atomic absorption type. While there is considerable difference in these two types of spectrographs, information available at this time indicates that neither has shown to be effective in diagnosis of hydraulic systems. The reason reported for this effectiveness is the inability of spectrometric techniques to "see" the large size particles which are generated in many wearing situations [6].

The development of the Ferrography technology offered some new hope for diagnosing hydraulic systems. Somewhat like the spectrograph, the Ferrograph evaluates the characteristics of the wear debris to appraise the magnitude of any wear process present in the system. The spectrograph, however, can only measure the *"parts per million"* of the various elements (such as iron, copper, lead, etc.), while the Ferrograph considers full particle morphological

analysis – size, shape, surface texture, homogeneity, color, bi-refringence, etc. – as well as debris concentration.

The Ferrograph system is a unique analytical system designed especially for the appraisal of wear processes. The system in use at the Fluid Power Research Center consists of a D. R. (Direct Reading) Ferrograph, a Slide Ferrograph, a Ferrogram Reader, and a Bichromatic Microscope with photographic accessories. The Ferrograph utilizes a specially developed magnet which generates an ultra-high gradient field near its poles to force the wear particles to precipitate from the vehicle fluid. In the case of the D. R. unit, the particles are collected from the fluid flowing axially along a glass tube, while the Slide Ferrograph uses an inclined chemically treated slide as a substrate upon which the attracted wear particles are collected. A carefully controlled pressure head on the fluid sample is used to continually pass a low velcoity stream through the glass precipitator tube or longitudinally along the slide. The slide or Ferrogram, as it is called, is analyzed using both a densitometer (Ferrogram Reader) and the Bichromatic Microscope. The densitometer is used to appraise the amount of wear debris collected, while the microscope permits the identification of particle material and wear modes.

Surface deterioration between moving parts is a fact of life in a hydraulic system. The modes and rate of wear which are active at any given time in such a system will depend upon the materials utilized, the duty cycle, loading on the moving parts, the operating environment, and the anti-wear deterrents used. When a hydraulic system or component is operated, literally billions of wear particles enter the system fluid. The size, shape, color, etc. of these particles depend upon the dominant wear modes and the material involved. However, particle sizes normally range from a few nanometers to several micrometres. Usually, the highly stressed wearing parts of a hydraulic component are made of steel; hence, wear debris sloughed from these parts exhibit strong magnetic characteristics. However, strange as it may seem, particles normally unaffected by magnetic fields become magnetically active when they have been involved in a wear process such as cold working, ploughing, gouging, cutting, etc. It is these magnetic properties exhibited by wear associated debris upon which the Ferrographic system capitalizes.

The information obtained from the Ferrographic system is both quantitative and qualitative. The D. R. Ferrograph provides a scaled value for the debris density at a position near the entrance of the precipitator tube (influent) and a value near the exit (effluent). The Ferrogram Reader gives the optical density of the wear debris deposited at various positions along the Ferrogram. Theoretically, the distribution of the density readings is a function of particle size, since larger particles will tend to be deposited quicker than smaller ones. The location of a particular density reading taken from a Ferrogram is always given in terms of the millimetre distance from the exit end of the slide. For example, a density reading indicated at 54 mm. was taken at a point 54 millimetres from the end where the fluid exits the slide. The information gained with the Bichromatic Microscope is qualitative in nature, since the operator reports only what his experience and training enable him to identify and relate. The special lighting features of the unique microscope permit the investigator to differentiate particle shape, patterns in the deposit, structural appearance, free metal, non-metal, oxides, color, and surface texture. Thus, it is possible to obtain a clear picture of prevalent wear modes as well as unusual insight as to the chemical nature of the environment where the particles were generated.

From the preceding description of the Ferrographic fluids analysis technique, it should be obvious why it has generated much enthusiasm and interest throughout the fluid power industry. Test results at this point certainly indicate that heavy wear is accompanied by high density readings from the Ferrographic system. Also, contaminant wear is a process which produces cutting type particles in fluid power pumps. These particles actually resemble the ribbons generated from a lathe during a machining operation. Although it is necessary to collect more information before taking a firm position, there is every indication at this point that the Ferrographic system will make a very useful diagnostic tool.

CHAPTER IV

TEST EFFORT AND RESULTS

With the guidance and consent of the program sponsors, the initial test effort was directed toward fluid power pumps. Each sponsor was requested to submit two each of two different pumps which they would like to see tested. From this request, a total of 12 pumps were received. Since there was no information concerning the wear debris generated from a hydraulic pump when operated at rated conditions in clean fluid, each pump was subjected to an extended break-in period before exposing it to contaminant in order to establish a normal wear base line. After the break-in evaluation, the pumps were subjected to a contaminant sensitivity test [4, 5]. In most cases, this was the standard contaminant sensitivity test with only changes in speed or pressure, although in two cases the contaminant level was altered from 300 mg/litre to 75 mg/litre.

Of the 12 pumps received for use in the program, ten were subjected to a complete test sequence (extended break-in through contaminant tests). One pump failed during break-in testing. Observations of this pump revealed that the front bearing apparently failed, resulting in a secondary failure in the shaft seal. The other pump which was not tested exhibited a malfunction in its basic operation and has been returned for repair. A complete summary of the test data and analysis is contained in the appendices to this report section.

BREAK-IN TESTS

The purpose of conducting the extended break-in tests was twofold. First of all, it was deemed necessary to obtain Ferrographic information relative to a clean fluid situation in order to have a base line for contaminant wear. In addition, it was desired to discover how

much wear debris is generated by a pump during the first few hours of operation and how soon does the high generation period subside. The break-in tests were conducted in a manner similar to that specified in the pump contaminant sensitivity test procedure [5]. That is, the pump was run at 25% rated pressure for 15 minutes; then, the pressure was increased to 50% rated and 75% rated pressure for 15 minutes, respectively. When the pump was brought up to full rated pressure, it was allowed to remain there for periods from one hour to eight hours while samples were taken for Ferrographic analysis. Since the generation characteristics of a pump were unknown at the start of these tests, it was impossible to predict the required length of the test. It was assumed that eight hours was certainly a maximum which would be necessary to establish a trend. However, to run each pump for eight hours would require considerable test time. Therefore, some pumps were tested for a shorter period.

In addition to the complete summary of data which can be found in the appendix, Figs. 4-1, 4-2, 4-3, and 4-4 show some of the break-in test data. These figures illustrate curves of the Ferrographic density reading at 54 millimetres, D54, versus the time duration of the test. The 54 mm location was chosen because it reflected the best correlation with the concentration and particle size of the wear debris generated. Fig. 4-1 illustrates the effect of pressure on two pumps from the same manufacturer, while Fig. 4-2 shows what occurred when two different speeds were used on two pumps from the same manufacturer. From Figs. 4-1 and 4-2, it would be simple to conclude that a pump will generate more debris and take longer to break in at high pressure but higher speeds have little effect. However, when Fig. 4-3 and Fig. 4-4 are reviewed, this conclusion becomes somewhat premature. Fig. 4-3 shows the Ferrographic analysis of two identical pumps (same part number) run at the same conditions. Fig. 4-4 illustrates the same information using a pump different from that of Fig. 4-3. It can be seen from these latter two figures that there is a large variation between pumps; therefore, the differences shown in Figs. 4-1 and 4-2 could have been in the pumps themselves instead of the different operating conditions.

In considering the length of time that it takes for a pump to reach a normal (low level) wear rate, it can be seen from the curves shown in Figs. 4-1 – 4-4 and the data in the appendices that, in all cases, the rate of debris generation was greatly reduced after one hour of running time at

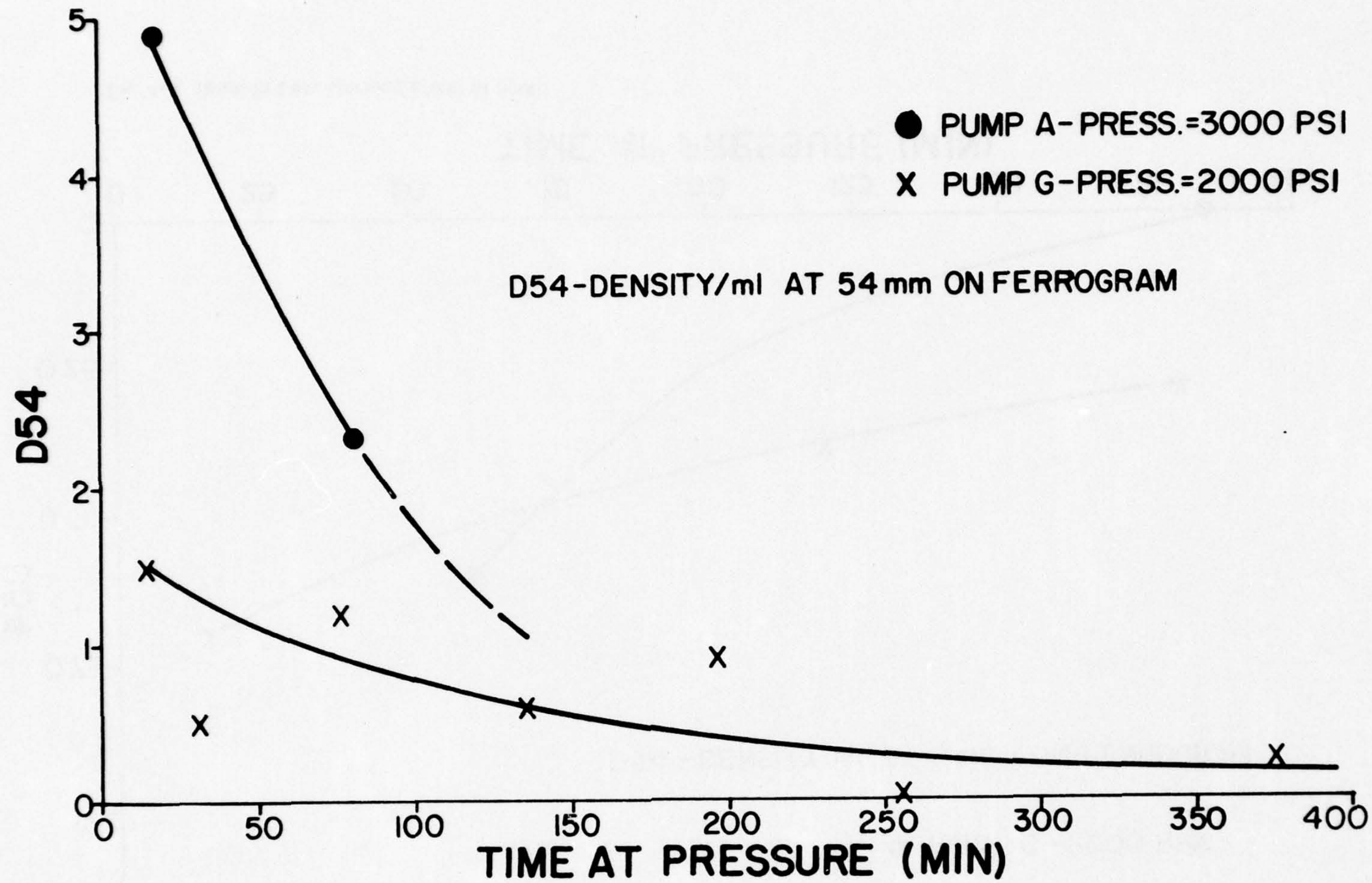

Fig. 4-1. Break-In Data Showing the Effect of Pressure.

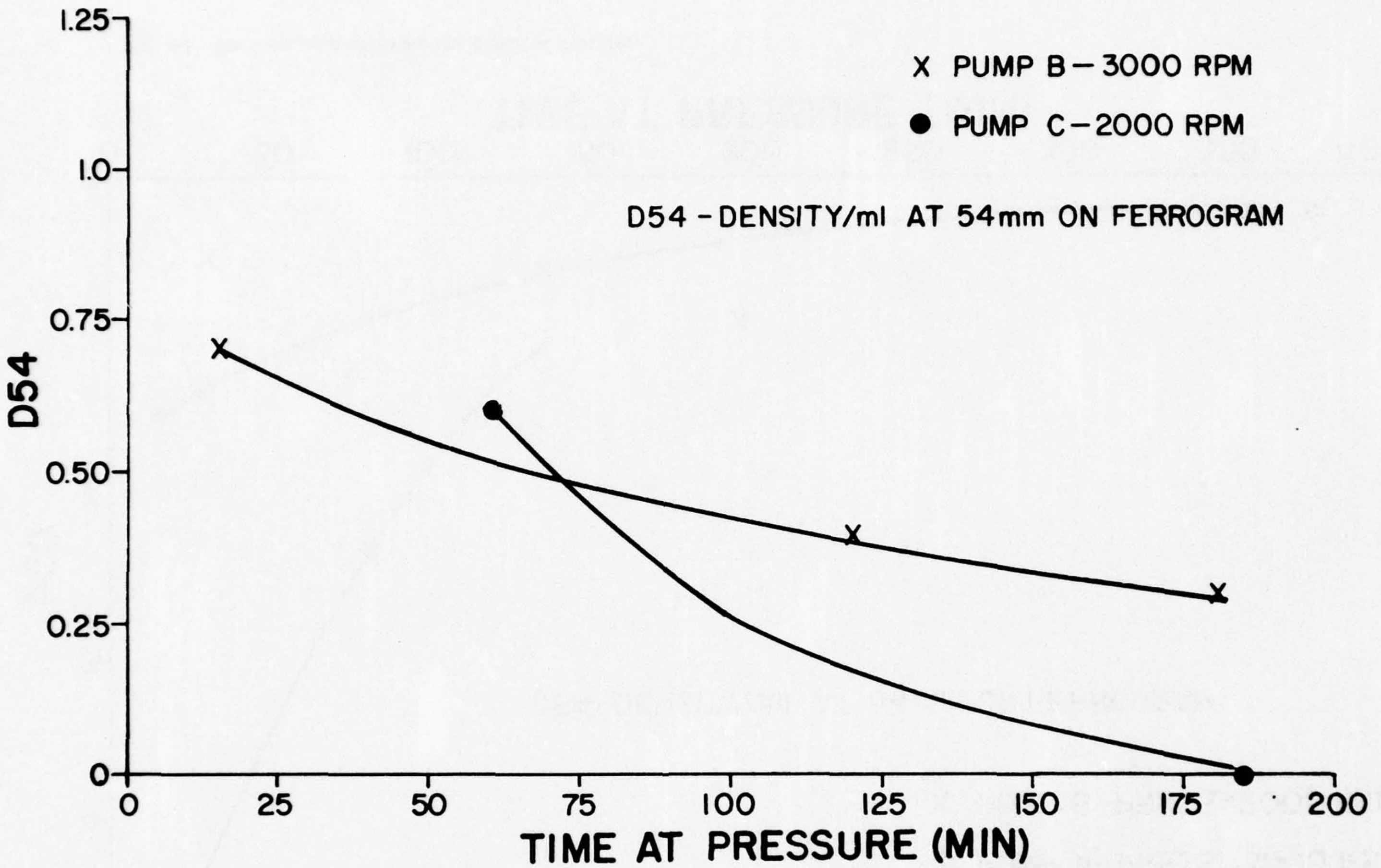

Fig. 4-2. Break-In Data Showing Effect of Speed.

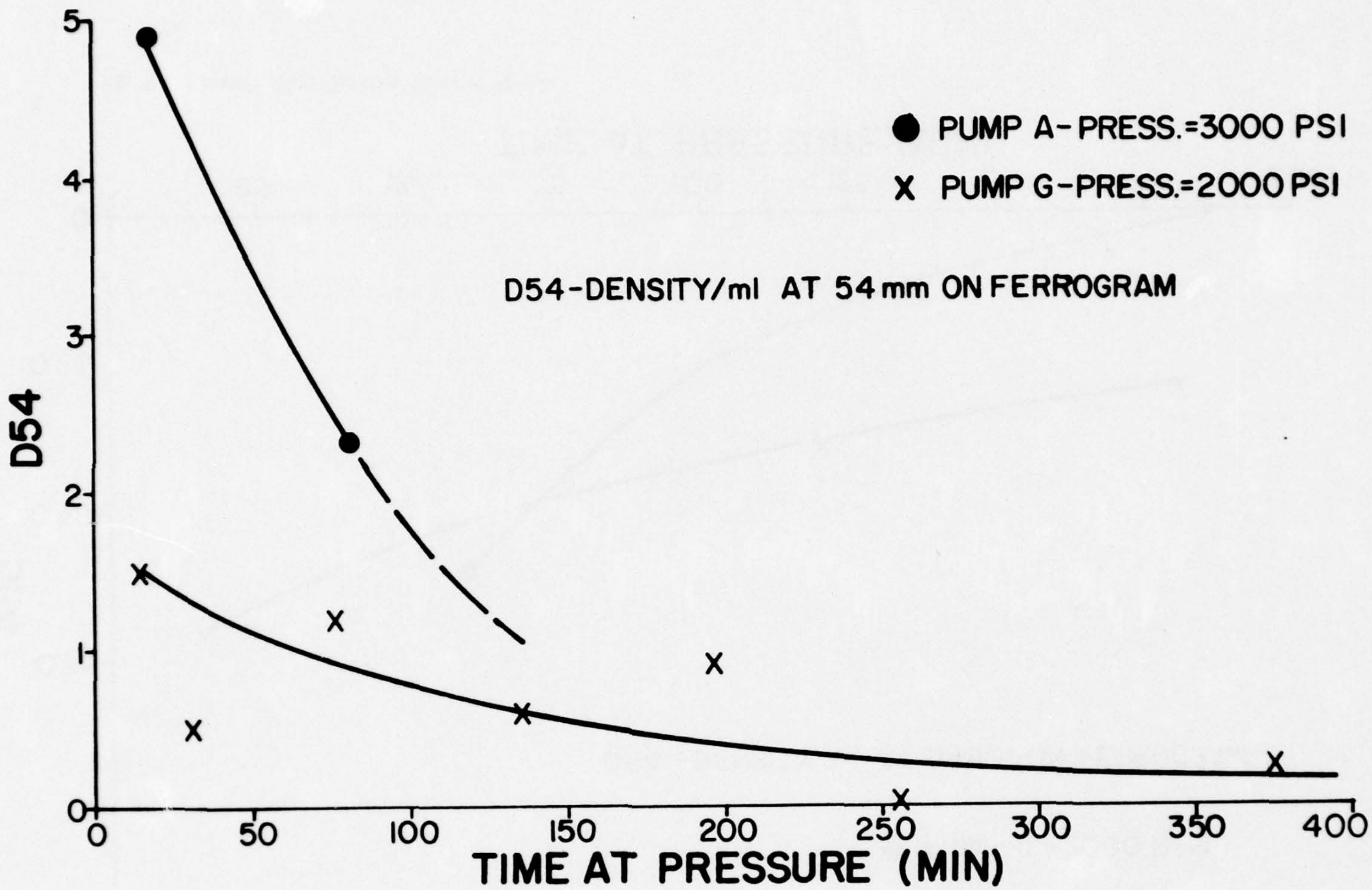

Fig. 4-1. Break-In Data Showing the Effect of Pressure.

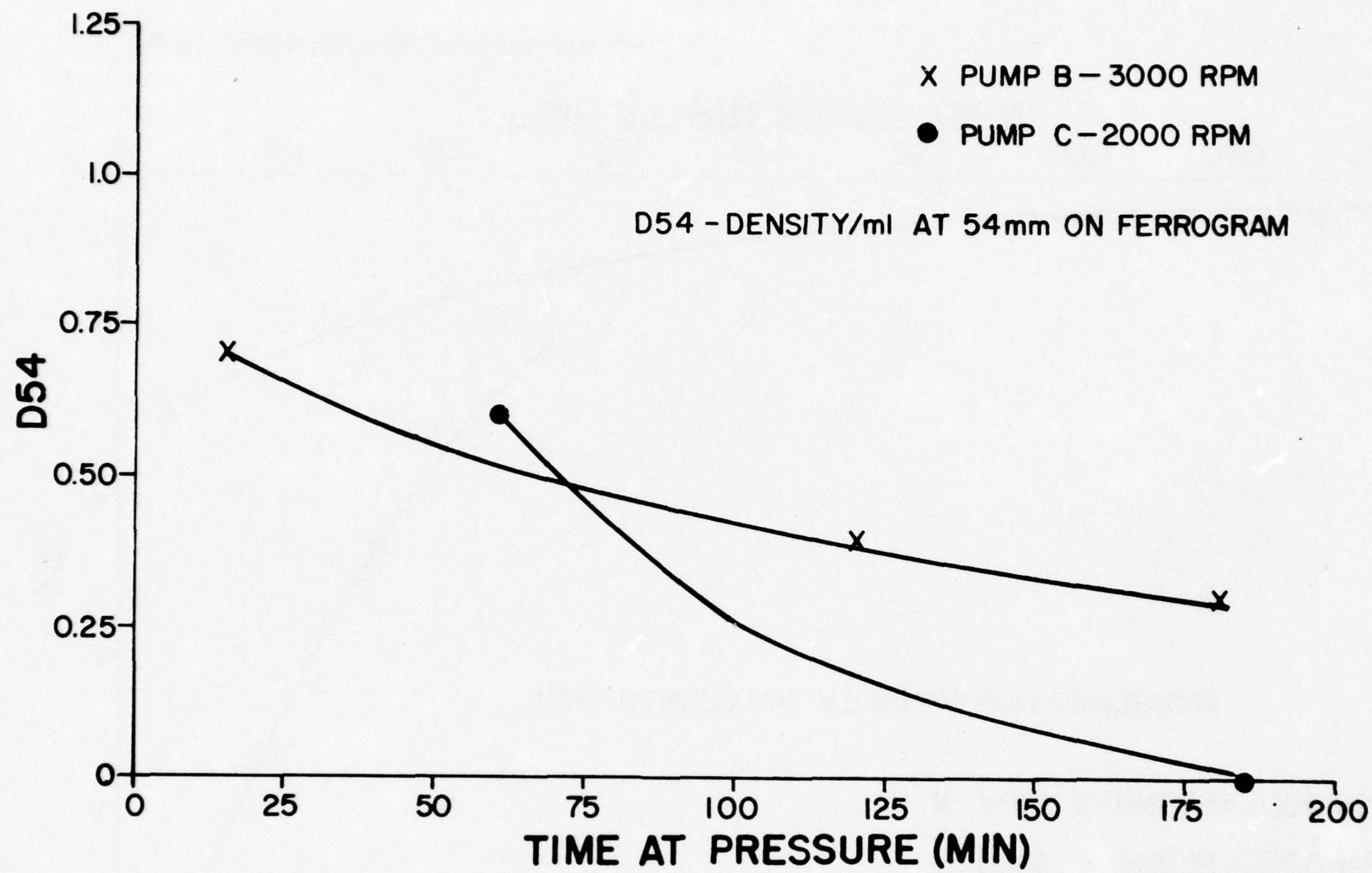

Fig. 4-2. Break-In Data Showing Effect of Speed.

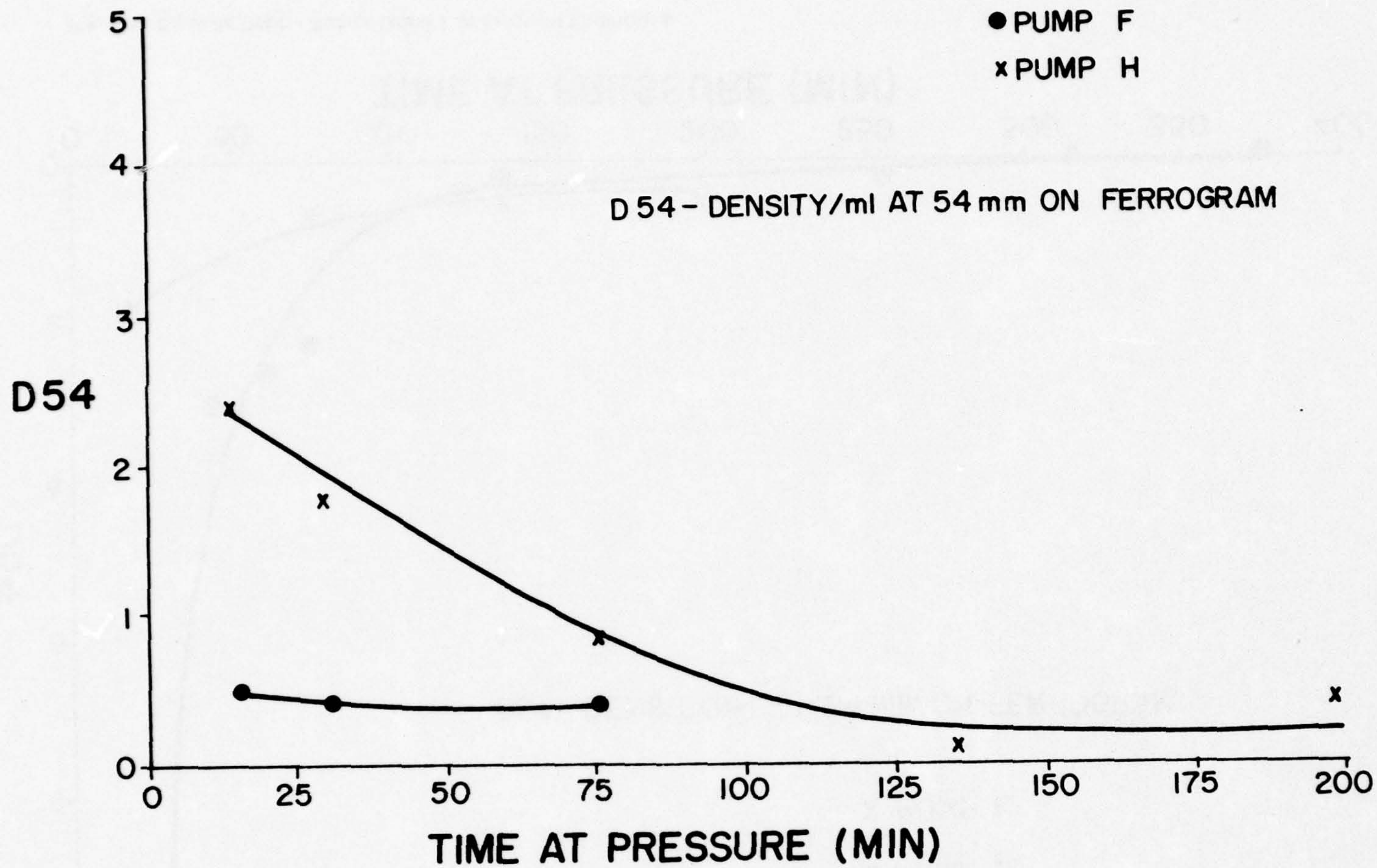

Fig. 4-3. Break-In Data – Pumps F and H at Identical Conditions.

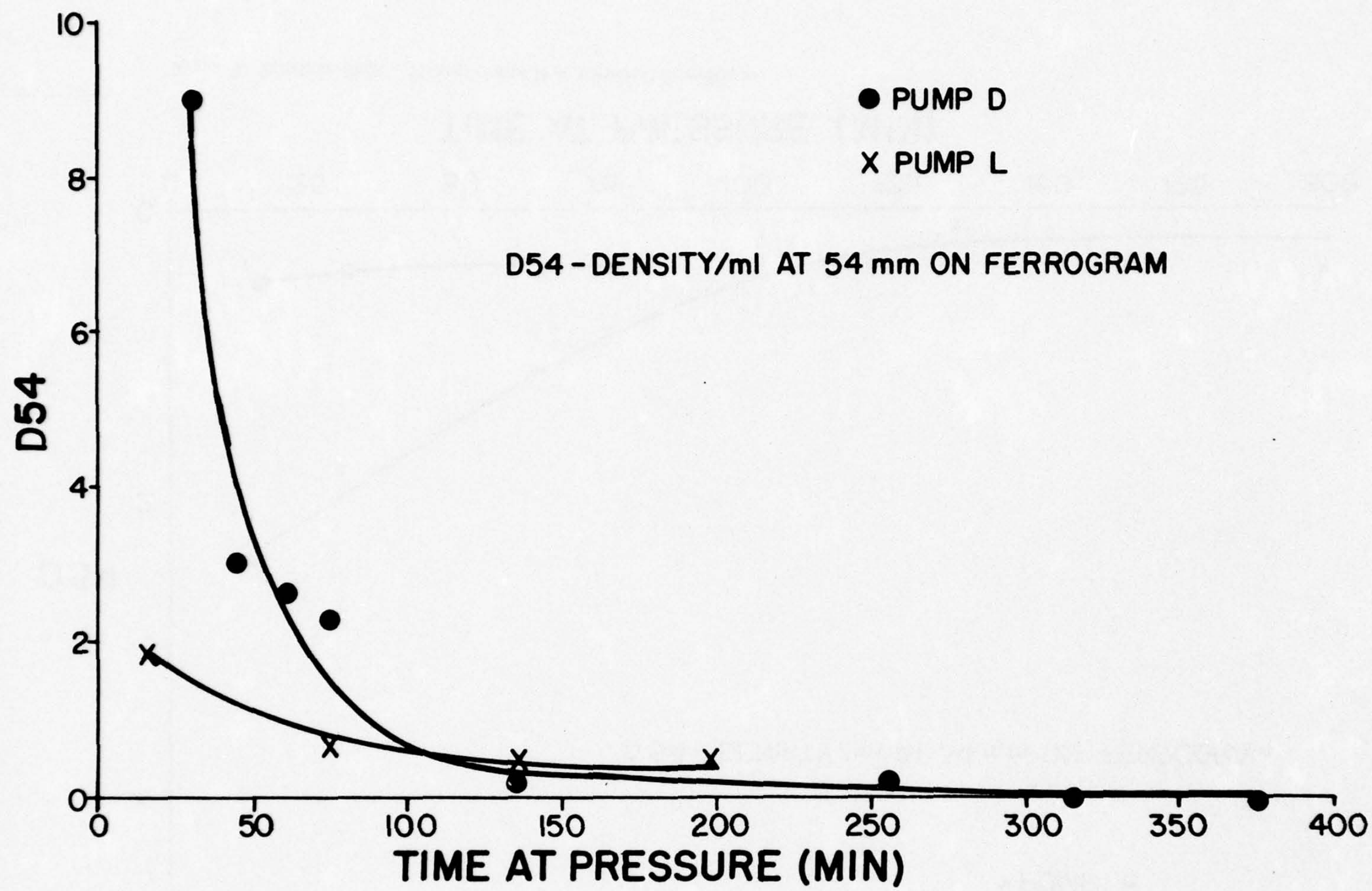

Fig. 4-4. Break-In Data – Pumps D and L at Identical Conditions.

full pressure. Since the samples were extracted immediately downstream of the pump to collect the wear being generated and the clean-up filters were being used during the break-in tests, the amount of debris collected on a Ferrograph slide is relative to the generation rate existing at the time of sampling. In the opinion of the authors of this report section, the break-in data collected show conclusively that a longer break-in period would have minimal effect upon the results of most testing efforts.

CONTAMINANT TESTS

Once a pump had been subjected to the break-in tests, the contaminant tests were initiated. The objectives of these contaminant tests were to evaluate the contaminant sensitivity of the pump submitted for testing and to evaluate the Ferrographic characteristics of the pump when subjected to contaminant wear. The flow degradation signatures normally associated with a standard contaminant sensitivity test [5] for the ten pumps tested are shown in Figs. 4-5, 4-6, and 4-7. All of the pumps were tested at 300 mg/litre with the exception of pumps designated L and M. After eight standard tests, it was decided to see if the amount of wear debris was sufficient to evaluate the contaminant sensitivity of a pump without requiring the high contaminant concentrations used for performance degradation detection.

The Ferrographic density readings for some *"typical"* pumps which had been tested are shown in Figs. 4-8, 4-9, 4-10, and 4-11. In Fig. 4-8, the two pumps are identical (same manufacturer, etc.), but Pump A was run at *3000 PSI outlet* pressure, while Pump G was tested at 2000 PSI. In addition, the two pumps shown in Fig. 4-9 are identical pumps which were tested at different speeds. Pump B was run at 3000 RPM, while Pump C was operated at 2000 RPM during the contaminant test. The two curves shown in Fig. 4-10 represent data from identical tests on identical pumps (Pumps E and J). The remaining two pumps (D and H) are illustrated in Fig. 4-11. It can be seen from these curves that higher pressures tend to make a pump generate more debris. This is consistent with other contaminant sensitivity testing which showed that a given pump was more sensitive to contaminant at higher pressures. The tests run at different speeds indicated that the pump wore less at the higher speed. This may have to do with the hydrodynamic film formation, but a full analysis has not been made at this time.

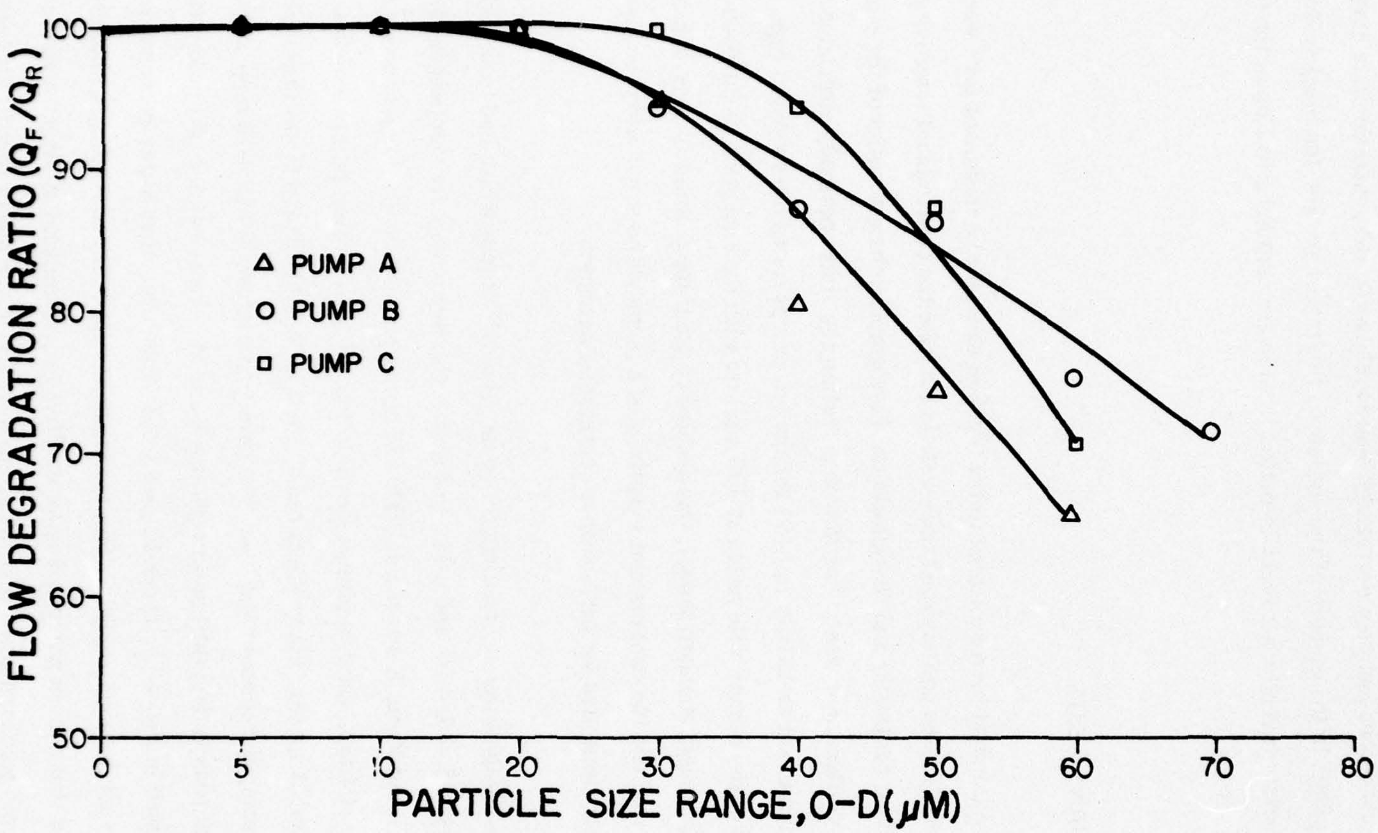

Fig. 4-5. Flow Degradation Signatures for Pumps A, B, and C. (Standard Test)

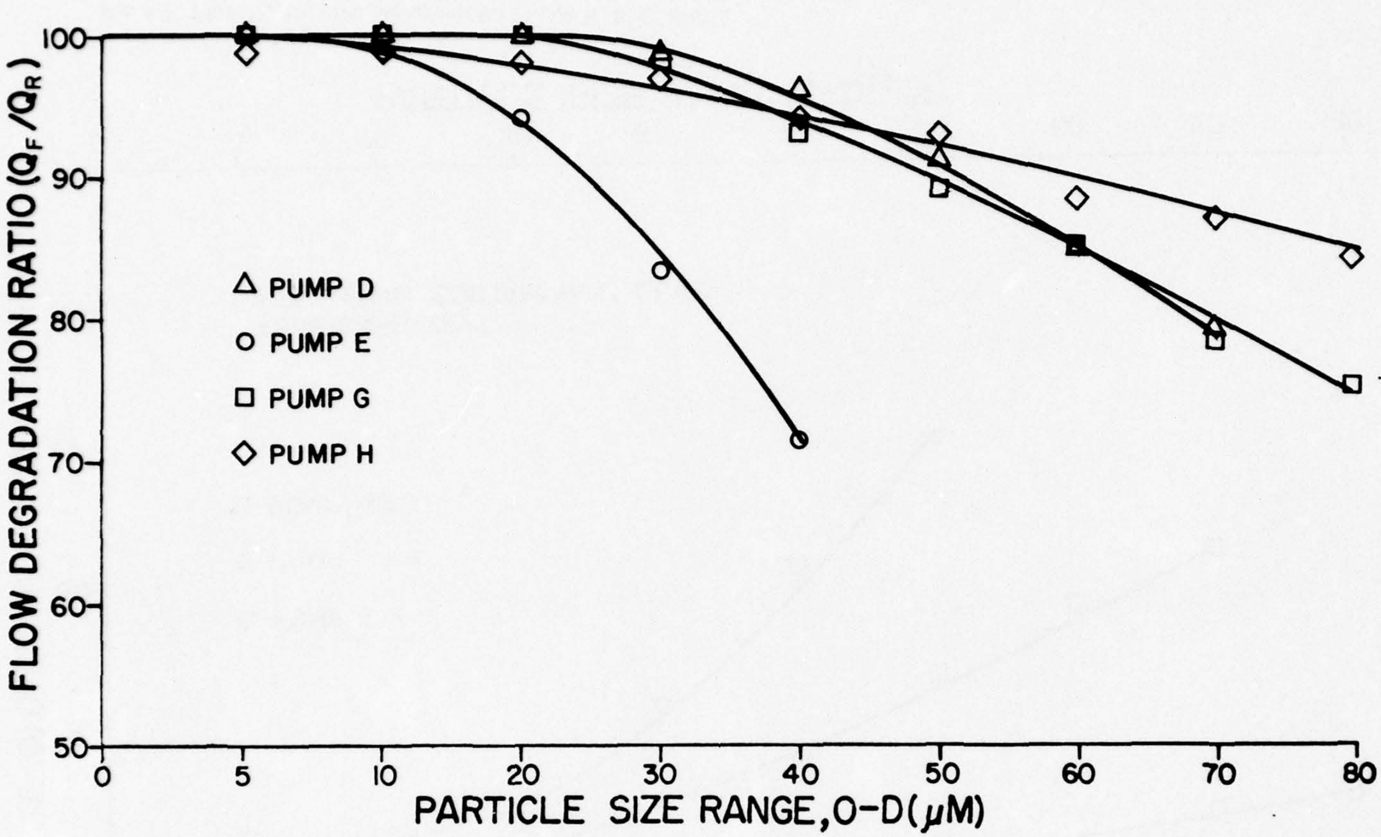

Fig. 4-6. Flow Degradation Signatures for Pumps J, L, and M.

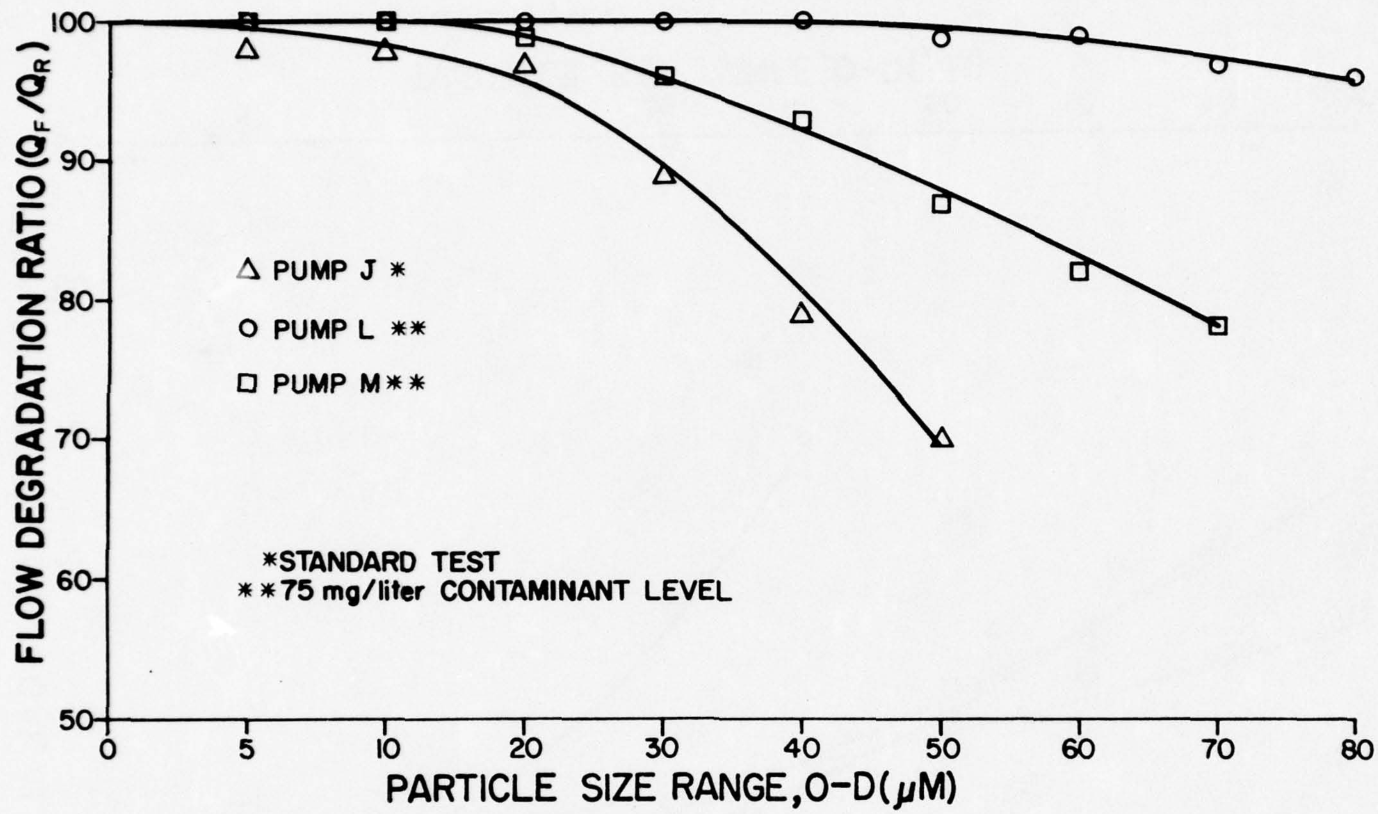

Fig. 4-7. Flow Degradation Signatures for Pumps D, E, G, and H.

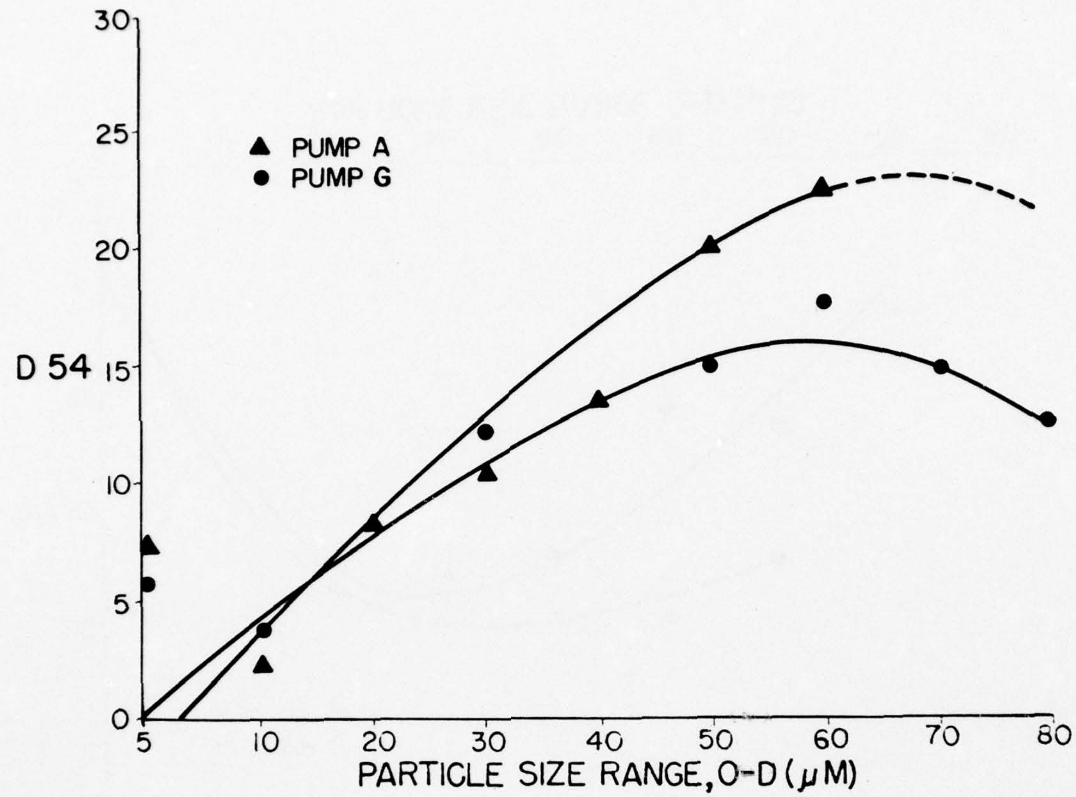

Fig. 4-8. Ferrographic Analysis of Pumps A and G.

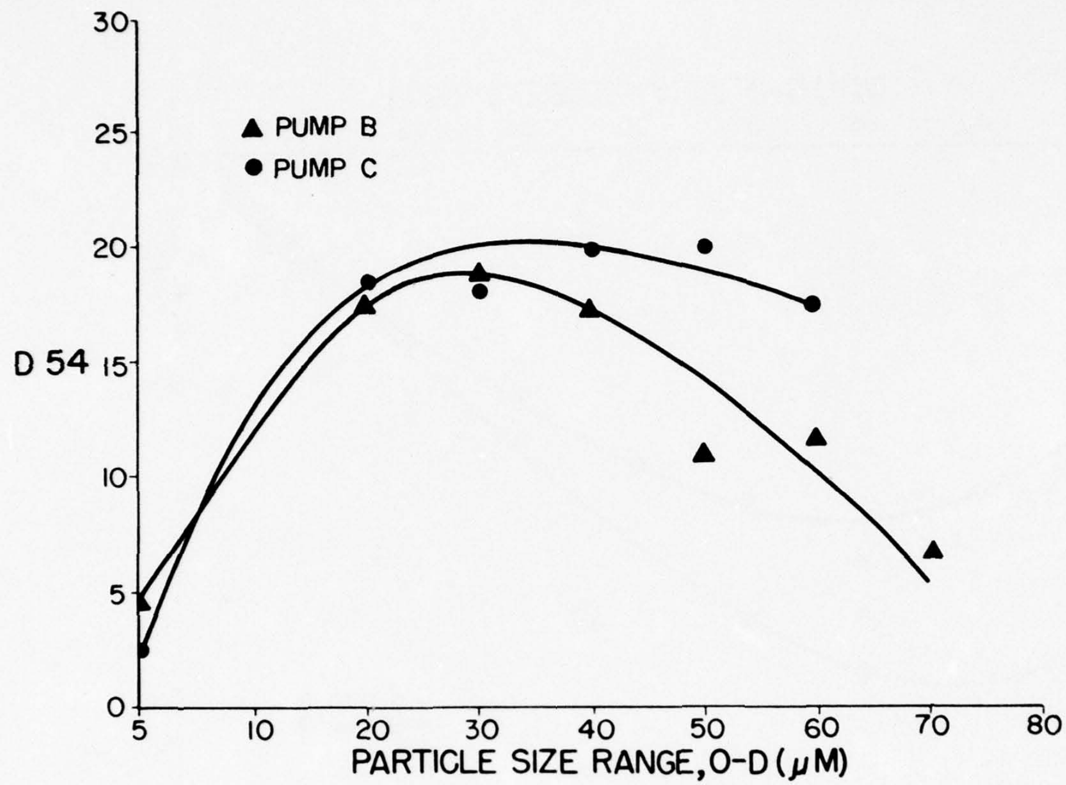

Fig. 4-9. Ferrographic Analysis of Pumps B and C.

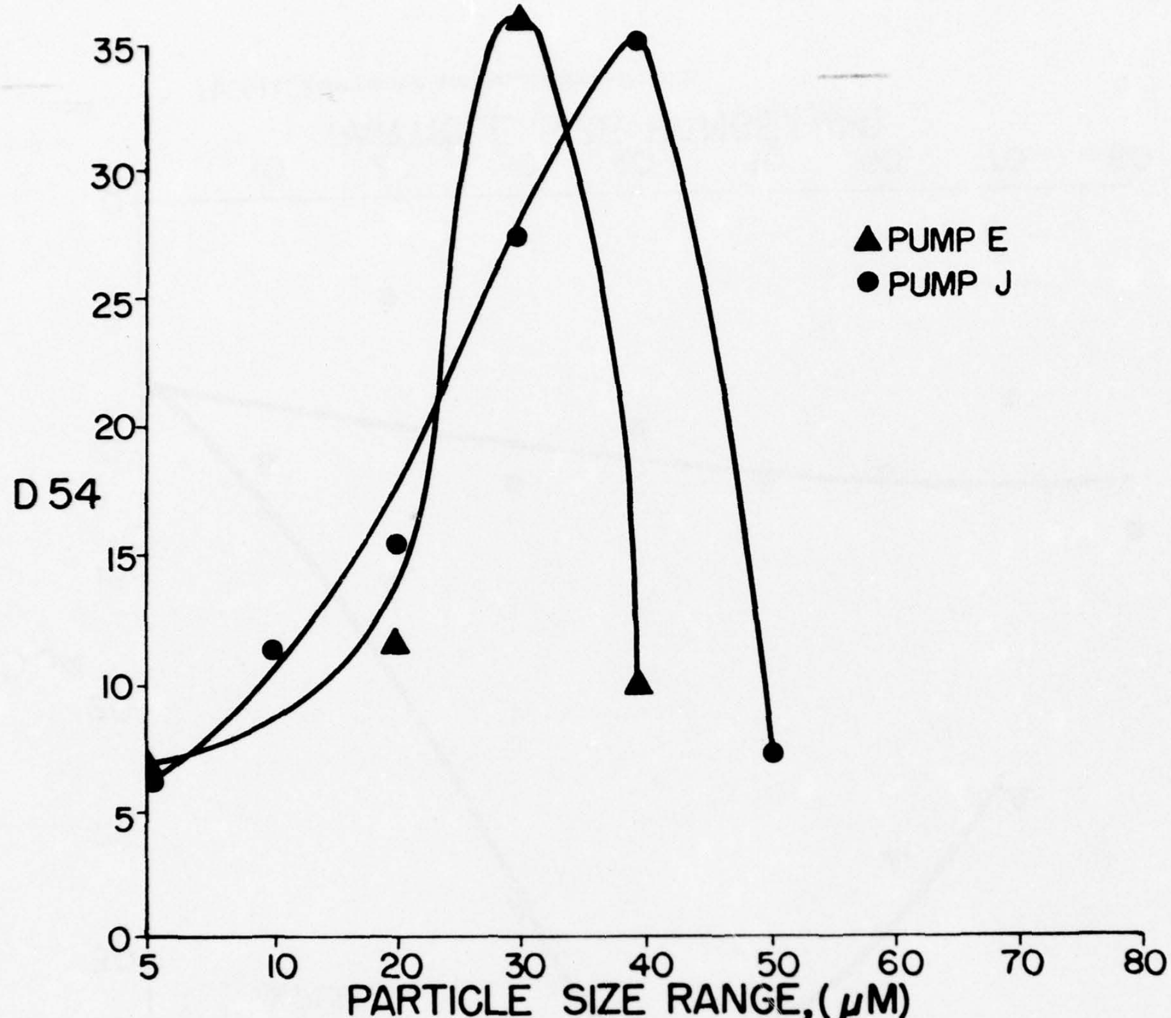

Fig. 4-10. Ferrographic Analysis of Pumps E and J.

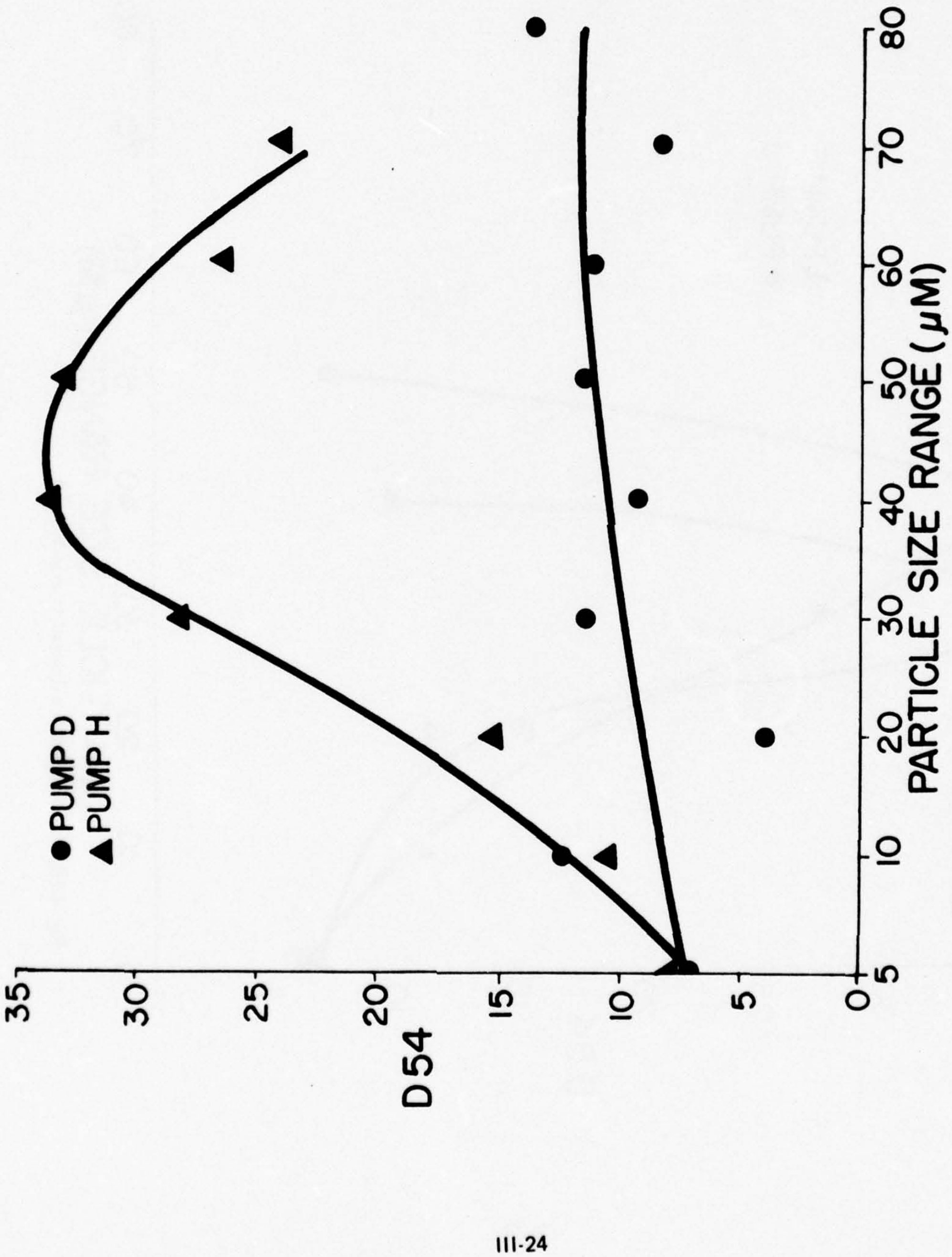

Fig. 4-11. Ferrographic Analysis of Pumps D and H.

CHAPTER V

INTERPRETATION OF RESULTS

Since it is anticipated that the project will be continued, only preliminary interpretations will be made. It is interesting to note that there seems to be a general trend in the Ferrographic density. That is, in all but one pump (H), the amount of wear debris increased up to a particular particle size and then decreased when larger particle sizes were injected. The full implication of this observation has not been totally explored. It may mean that a pump exhibits what has been termed a critical particle size. When this size is approached, the amount of wear and, therefore, the amount of wear debris will increase. However, subjecting the pump to sizes of larger than this critical size does not cause greater wear. In fact, if the number of particles in the critical size range decreases in the larger size range, then the wear may decrease, as shown by the debris analysis.

Another possible explanation for the trend observed from the Ferrographic density lies in the fact that all of the pumps tested are thought to be of the wear compensating design. That is, they include a pressure loaded wear plate which is designed to press against the sides of the gears and compensate for material loss in this location. Obviously, after a certain amount of wear has occurred, the pump will cease to compensate effectively. The trend being reflected by the data produced during this project may be revealing critical information concerning the action of wear-compensated hydraulic pumps.

From a theoretical standpoint, the amount of wear debris should be expected to decrease toward the end of the contaminant test. If the actual flow degradation is studied from the summary tables in the appendix, it can be seen that the loss of flow during each size exposure is fairly constant at greater particle sizes. For example, with Pump B, there is no degradation until the pump was exposed to 0-30 micrometre contaminant. The loss of flow was about one GPM for 0-30, 0-40, 0-50, 0-60, and 0-70. If the assumption is made that the leakage or

slip flow of the pump follows a cubic relationship with respect to some characteristic clearance (as many experts have assumed) [7], then the slip flow can be illustrated as shown in Fig. 5-1. Furthermore, assume that the actual slip flow before any contaminant exposure was 0.2 GPM, which represents a characteristic clearance of about 1.46. If by exposing this pump to 0-30 micrometre contaminant the slip flow increases by one GPM, then the characteristic clearance must have changed from 1.46 to 2.66 for an increase of 1.2. When the pump was exposed to 0-40, the slip flow increased by another one GPM. Therefore, the total slip flow changed from the 1.2 at the end of 0-30 exposure to 2.2 at the end of 0-40. However, the clearance only changed from 2.66 to 3.25 to accommodate the slip flow change. Thus, when exposed to 0-30, the characteristic clearance of the pump increased by 1.2; but, when exposed to 0-40, it increased by only 0.6. By studying Fig. 5-1, it can be seen that this trend continues as long as the flow degradation is constant.

Now, let us assume that the amount of wear debris generated by a pump is directly proportional to the change in this characteristic dimension. Therefore, since the flow loss increase was constant with contaminant size range for Pump B, we can conclude that the change in the characteristic clearance decreased with size. Thus, it should be expected that the wear debris generated when the pump was exposed to 0-70 would be less than that generated from the 0-30 exposure. By observing the D54 values in the summary table for Pump B, it will be seen that this is the case. In fact, most of the pumps tested revealed a tendency to reach a given flow loss value and remain fairly constant for subsequent contaminant injections. This could account for the peaking out trend observed in the Ferrographic density results. Of course, the pumps will compensate for some of the wear and thus will probably not exactly follow the preceding theory.

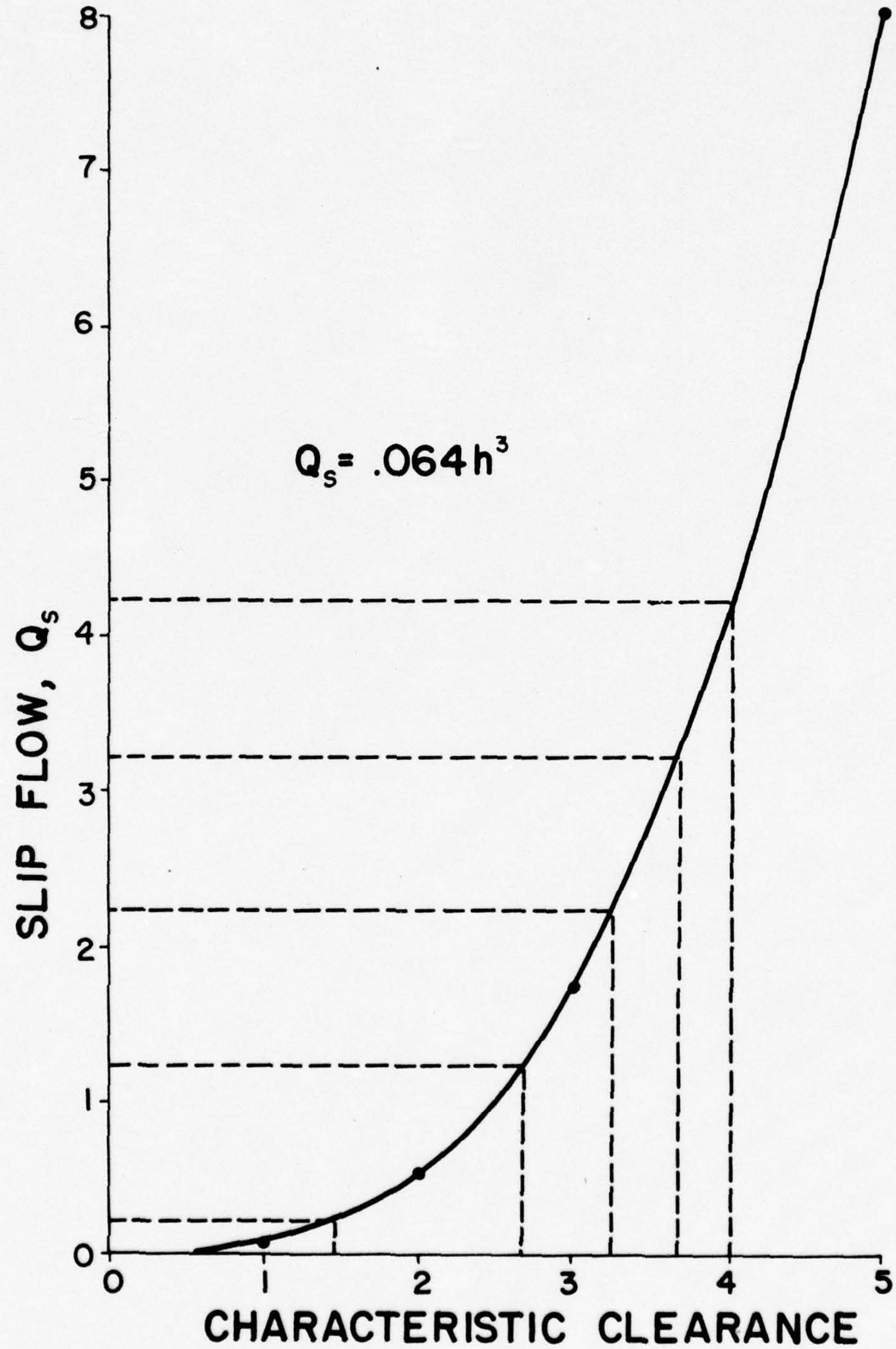

Fig. 5-1. Pump Slip Flow Vs. Characteristic Clearance.

CHAPTER VI

CONCLUSIONS

From the data presented and the interpretation, it must be concluded that Ferrographic analysis is a powerful technique for evaluating the wear characteristics of fluid power components. Since there must be more work accomplished in this area, only the foolhardy would venture conclusions of a concrete nature. However, it would appear that the characteristic clearance theory has considerable merit and could prove to be a major breakthrough in pump contaminant wear theory. In fact, further investigation into this phenomenon could provide the key to successful hydraulic system diagnostics that has never been possible in the past. The data obtained from the extended break-in tests would indicate that the pump is not completely broken in at the normal one-hour constant pressure break-in procedure. However, a great majority of the normal break-in process has been completed in this one hour, and the seat-in that is accomplished in another hour is probably of little value.

Without a doubt, it must be concluded that the evidence and information obtained thus far in the program would certainly dictate that the effort be continued. The analysis which can be performed on the Bichromatic Microscope offers promise of a more in-depth interpretation. The fact that the Ferrographic data can yield so much information relative to the wear characteristics of a pump would indicate that such rigorous results and insight can be obtained concerning the entire system.

CHAPTER VII

RECOMMENDATIONS

Based on inputs from the majority of the project sponsors, the progress made to date on the activity certainly indicates that it should be continued. Specifically, it has been suggested that tests be conducted on no more than five more pumps in order to fully define the characteristic clearance concept. The project should then consider the testing of hydraulic valves. This effort would be limited to directional and relief valves and would include no more than eight tests.

The primary thrust of the continuation of this project would be towards an intensive field system evaluation phase. The advice received from industrial sponsors indicates that most hydraulic systems operating in the field are not independent of other systems. That is, in many cases, the reservoir for the hydraulic system is actually the transmission of the vehicle. Therefore, the wear debris entrained in the system fluid could be generated from the gears and clutches of the transmission as well as the hydraulic components. In general, the implicit concensus of the industrial sponsorship is that the laboratory test phase of this project was effective in providing background information and confidence in the technique. However, due to the need to develop an effective field diagnostic technique, the field system evaluation should be initiated without delay. It is hoped that the contract monitors of the MERDC staff will concur with these recommendations.

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2. Seifert, W. W. and V. C. Westcott, *"A Method for the Study of Wear Particles in Lubrication Oil,"* **Wear Magazine**, Elsevier Sequoia, S.A , Vol. 21, pp. 27-42.

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4. *"Hydraulic Specification Program,"* Annual Report, December 1971, AD 737730, Prepared for U.S. Army Mobility Equipment Research and Development Center, Prepared by Fluid Power Research Center, Oklahoma State University, Stillwater, Oklahoma, December 1971.

5. Fitch, E. C., *"Component Contaminant Sensitivity – A Status Report on Pumps,"* Proc. of National Conference on Fluid Power, Vol. XXVIII, Philadelphia, Pennsylvania, 1974.

6. Westcott, V. C., and W. W. Seifert, *"Investigation of Iron Content of Lubricating Oil Using a Ferrograph and an Emission Spectrometer,"* **Wear Magazine**, Vol. 23, 1973, pp. 239-249.

7. Blackburn, J. F., G. Reethof, and J. L. Shearer, *"Fluid Power Control,"* The M.I.T. Press, The Massachusetts Institute of Technology, Cambridge, Massachusetts, 1969.

APPENDIX A

SUMMARY OF PUMP EXTENDED BREAK-IN TESTS

SUMMARY OF PUMP BREAK-IN DATA AND ANALYSIS

Pump #: A | Pump Type: Gear

Pump Speed: 2250 RPM | Fluid Temp.: 65 °C

Inlet Press: 5 PSI

Outlet Press. (PSI)	Q (GPM)	Time At Pressure (Min)	Sample Time (Min)	DRI	DRE	DEN	D54	D50	D30	D10
3000	38.0	18	18	43.1	10.7	9.9	4.9	4.6	1.9	2.3
	38.2	80	80	11.2	3.1	3.8	2.3	.9	1.4	.9

DRI: Influent Density per 1 cc. of sample by direct reading Densiometer.
DRE: Effluent density per 1 cc. of sample by direct reading Densiometer
DEN: Density per 1 cc. of sample at entry of Ferrogram
D54: Density per 1 cc. of sample at 54 mm. on Ferrogram

D50: As D54 except at 50 mm.
D30: As D54 except at 30 mm.
D10: As D54 except at 10 mm.

SUMMARY OF PUMP BREAK-IN DATA AND ANALYSIS

Pump #: B

Pump Type: Gear

Pump Speed: 3000 RPM

Fluid Temp.: 65 °C

Inlet Press: 5 PSI

Outlet Press. (PSI)	Q (GPM)	Time At Pressure (Min)	Sample Time (Min)	DRI	DRE	DEN	D54	D50	D30	D10
500	18.6	1	1	37.7	18.0	9.1	7.2	5.8	3.6	5.4
500	17.8	15	15	16.8	8.2	4.1	1.1	1.1	1.1	3.4
1000	17.7	15	15	11.2	4.5	2.6	.9	1.5	0.0	.7
1500	17.7	15	15	10.5	4.8	1.8	.4	0.0	1	2.1
2000	17.8	15	15	6.7	2.5	2.1	.7	.7	.7	1.3
2000	17.9	60	60	6.3	2.0					
2000	18.1	120	120	5.0	1.7	1.7	.4	.4	1.1	1.7
2000	17.9	180	180	3.5	1.3	1.7	.3	0.0	.2	.9

DRI: Influent Density per 1 cc. of sample by direct reading Densiometer.

DRE: Effluent density per 1 cc. of sample by direct reading Densiometer

DEN: Density per 1 cc. of sample at entry of Ferrogram

D54: Density per 1 cc. of sample at 54 mm. on Ferrogram

D50: As D54 except at 50 mm.

D30: As D54 except at 30 mm.

D10: As D54 except at 10 mm.

SUMMARY OF PUMP BREAK-IN DATA AND ANALYSIS

Pump #: C   Pump Type: Gear

Pump Speed: 2000 RPM   Fluid Temp.: 65 °C

Inlet Press: 5 PSI

Outlet Press. (PSI)	Q (GPM)	Time At Pressure (Min)	Sample Time (Min)	DRI	DRE	DEN	D54	D50	D30	D10
500	14.9	1	1	9.1	6.3	6.1	1.2	2.2	1.6	2.2
500	14.4	15	15	9.8	3.6	2.3	1.3	1.0	.3	1.1
1000	13.5	15	15	10.9	3.6	2.6	1.1	1.1	.9	1.2
1500	12.8	15	15	19.8	4.9	3.8	2.4	2.0	.9	.8
2000	11.5	65	65	6.5	1.7	.9	.6	0.0	0.0	1.0
2000	11.5	125	125	1.8	1.5	2.3	0.0	0.0	0.0	0.0
2000	11.6	185	185	3.1	1.8	.3	0.0	0.0	0.0	0.0

DRI: Influent Density per 1 cc. of sample by direct reading Densiometer.

DRE: Effluent density per 1 cc. of sample by direct reading Densiometer

DEN: Density per 1 cc. of sample at entry of Ferrogram

D54: Density per 1 cc. of sample at 54 mm. on Ferrogram

D50: As D54 except at 50 mm.

D30: As D54 except at 30 mm.

D10: As D54 except at 10 mm.

SUMMARY OF PUMP BREAK-IN DATA AND ANALYSIS

Pump #: D   Pump Type: Gear

Pump Speed: 2000 RPM   Fluid Temp.: 65 °C

Inlet Press: 5 PSI

Outlet Press. (PSI)	Q (GPM)	Time At Pressure (Min)	Sample Time (Min)	DRI	DRE	DEN	D54	D50	D30	D10
2000	18.2	30	30	54.4	19.3	8.5	9.0	6.6	3.6	3.8
2000	18.2	45	45	32.7	9.6	2.8	3.0	3.0	1.4	1.7
2000	18.2	60	60	20.7	8.6	1.8	2.6	.9	.2	1.6
2000	18.2	75	75	9.6	4.0	2.3	2.3	1.4	.6	2.3
2000	18.2	135	135	9.5	2.2	.8	.2	0.0	.1	1.4
2000	18.1	255	255	11.9	2.2	.5	.2	0.0	.1	.2
2000	18.2	315	315	2.3	1.7	.3	0.0	0.0	0.0	0.0
2000	18.1	375	375	4.8	1.3	.7	0.0	0.0	.8	.3
2000	18.3	435	435	10.5	2.1	.9	.1	0.0	0.0	0.0

DRI: Influent Density per 1 cc. of sample by direct reading Densiometer.

DRE: Effluent density per 1 cc. of sample by direct reading Densiometer

DEN: Density per 1 cc. of sample at entry of Ferrogram

D54: Density per 1 cc. of sample at 54 mm. on Ferrogram

D50: As D54 except at 50 mm.

D30: As D54 except at 30 mm.

D10: As D54 except at 10 mm.

SUMMARY OF PUMP BREAK-IN DATA AND ANALYSIS

Pump #: E

Pump Type: Gear

Pump Speed: 1800 RPM

Fluid Temp.: 65 °C

Inlet Press: 5 PSI

Outlet Press. (PSI)	Q (GPM)	Time At Pressure (Min)	Sample Time (Min)	DRI	DRE	DEN	D54	D50	D30	D10
500	21.8	15	15	28.5	6.6	4.8	4.4	3.9	2.4	3.2
1000	20.8	15	15	41.5	8.3	5.7	5.8	5.9	3.1	4.5
1500	19.5	15	15	9.1	2.8	1.9	1.8	2	.3	2.4
2000	18.8	15	15	29.9	8.7	3.6	2.6	3.1	.6	.7
2000	18.8	75	75	7.1	4.4	.6	.3	.2	.3	1.8
2000	18.8	135	135							
2000	19.2	195	195	6.5	2.2	1.6	.7	.2	0.0	0.0

DRI: Influent Density per 1 cc. of sample by direct reading Densiometer.

DRE: Effluent density per 1 cc. of sample by direct reading Densiometer

DEN: Density per 1 cc. of sample at entry of Ferrogram

D54: Density per 1 cc. of sample at 54 mm. on Ferrogram

D50: As D54 except at 50 mm.

D30: As D54 except at 30 mm.

D10: As D54 except at 10 mm.

SUMMARY OF PUMP BREAK-IN DATA AND ANALYSIS

Pump #: F Pump Type: Gear

Pump Speed: 2000 RPM Fluid Temp.: 65 °C

Inlet Press: 5 PSI

Outlet Press. (PSI)	Q (GPM)	Time At Pressure (Min)	Sample Time (Min)	DRI	DRE	DEN	D54	D50	D30	D10
500	33.4	15	15	15.2	5.3	2.3	2.3	1.6	1.0	.9
1000	32.8	15	15	16.7	5.3	4.2	2.9	1.5	.2	1.9
1500	32.2	15	15	7.1	3.3	1.8	.9	.6	.3	1.5
2000	31.6	15	15	11.0	4.6	1.8	.5	.6	1.0	1.9
2000	31.5	30	30	7.1	2.9	1.7	.4	.1	.3	1.0
2000	31.5	75	75	2.8	1.7	1.0	.4	.2	.3	1.7
2000	31.5	135	135							

DRI: Influent Density per 1 cc. of sample by direct reading Densiometer.

DRE: Effluent density per 1 cc. of sample by direct reading Densiometer

DEN: Density per 1 cc. of sample at entry of Ferrogram

D54: Density per 1 cc. of sample at 54 mm. on Ferrogram

D50: As D54 except at 50 mm.

D30: As D54 except at 30 mm.

D10: As D54 except at 10 mm.

SUMMARY OF PUMP BREAK-IN DATA AND ANALYSIS

Pump #: G
Pump Type: Gear
Pump Speed: 2000 RPM
Fluid Temp.: 65 °C
Inlet Press: 5 PSI

Outlet Press. (PSI)	Q (GPM)	Time At Pressure (Min)	Sample Time (Min)	DRI	DRE	DEN	D54	D50	D30	D10
500	24.1	15	15	12.1	6.2	2.5	.9	2.4	.2	1.4
1000	24.1	15	15	9.9	7.4					
1500	23.6	15	15	10.0	7.0	2.4	.2	.4	3.0	6.3
2000	22.8	15	15	6.9	8.1	2.3	1.5	.2	0.0	0.0
2000	23.0	30	30	3.7	2.3	1.1	.5	0.0	.1	1.9
2000	23.1	75	75	6.0	4.4	2.3	1.2	.6	1.5	1.3
2000	23.1	135	135	4.9	1.3	1.4	.6	.5	1.5	1.6
2000	23.1	195	195	5.5	1.7	.2	.9	.5	.5	.9
2000	23.1	255	255	1.3	0.6	3.7	0.0	0.0	0.0	0.0
2000	22.6	315	315	2.0	1.7	2.5	2.3	2.0	.7	1.1
2000	23.0	375	375	2.0	1.3	2.0	.2	.2	.5	7.9
2000	23.1	435	435	3.0	1.2	1.3	2.3	1.1	9.2	3.0

DRI: Influent Density per 1 cc. of sample by direct reading Densiometer.
DRE: Effluent density per 1 cc. of sample by direct reading Densiometer
DEN: Density per 1 cc. of sample at entry of Ferrogram
D54: Density per 1 cc. of sample at 54 mm. on Ferrogram
D50: As D54 except at 50 mm.
D30: As D54 except at 30 mm.
D10: As D54 except at 10 mm.

SUMMARY OF PUMP BREAK-IN DATA AND ANALYSIS

Pump #: H   Pump Type: Gear

Pump Speed: 1800 RPM   Fluid Temp.: 65 °C

Inlet Press: 5 PSI

Outlet Press. (PSI)	Q (GPM)	Time At Pressure (Min)	Sample Time (Min)	DRI	DRE	DEN	D54	D50	D30	D10
500	45.5	15	15	6.4	2.8	3.9	1.8	1.3	.4	.5
1000	44.0	15	15	5.2	3.0	1.5	.8	1.2	1.4	1.6
1500	43.0	15	15	2.7	1.4	2.1	0	.3	.7	1.0
2000	41.6	15	15	6.8	6.1	3.7	2.4	1.3	3.0	7.3
2000	41.3	30	30	4.1	1.6	1.9	1.8	1.4	2.1	1.6
2000	41.4	75	75	4.4	0.2	1.5	.8	.6	.4	.6
2000	41.5	135	135	2.5	1.6	.8	.1	0.0	0.0	0.0
2000	41.4	195	195	5.7	8.6	1.0	.4	.9	.2	1.0

DRI: Influent Density per 1 cc. of sample by direct reading Densiometer.
DRE: Effluent density per 1 cc. of sample by direct reading Densiometer
DEN: Density per 1 cc. of sample at entry of Ferrogram
D54: Density per 1 cc. of sample at 54 mm. on Ferrogram

D50: As D54 except at 50 mm.
D30: As D54 except at 30 mm.
D10: As D54 except at 10 mm.

SUMMARY OF PUMP BREAK-IN DATA AND ANALYSIS

Pump #: J Pump Type: Gear

Pump Speed: 2000 RPM Fluid Temp.: 65 °C

Inlet Press: 5 PSI

Outlet Press. (PSI)	Q (GPM)	Time At Pressure (Min)	Sample Time (Min)	DRI	DRE	DEN	D54	D50	D30	D10
500	27	15	15	12.2	5.1	5.1	5.6	3.8	3.7	2.3
1000	26.6	15	15	8.7	5.5	2.3	2.4	1.6	.7	1.0
1500	25.9	15	15	5.6	3.8	2.1	1.0	.9	2.4	1.0
2000	25.9	15	15	4.6	1.8	.8	.4	.7	.5	.2
2000	25.9	30	30	1.7	1.7	2.4	.9	.6	1.8	1.0
2000	25.9	135	135	1.3	1.3	.7	.9	.6	0.0	.9
2000	25.9	195	195	1.4	2.1	1.8	1.5	.8	1.3	.4
2000	25.9	255	255	1.7	12.9	2.1	.4	.2	.4	0.0

DRI: Influent Density per 1 cc. of sample by direct reading Densiometer.

DRE: Effluent density per 1 cc. of sample by direct reading Densiometer

DEN: Density per 1 cc. of sample at entry of Ferrogram

D54: Density per 1 cc. of sample at 54 mm. on Ferrogram

D50: As D54 except at 50 mm.

D30: As D54 except at 30 mm.

D10: As D54 except at 10 mm.

SUMMARY OF PUMP BREAK-IN DATA AND ANALYSIS

Pump #: L Pump Type: Gear

Pump Speed: 2000 RPM Fluid Temp.: 65 °C

Inlet Press: 5 PSI

Outlet Press. (PSI)	Q (GPM)	Time At Pressure (Min)	Sample Time (Min)	DRI	DRE	DEN	D54	D50	D30	D10
500	19.1	15	15	3.8	2.9	1.3	1.3	1.1	1.9	1.3
1000	19.0	15	15	1.6	1.4	2.6	.5	1.7	1.7	1.3
1500	20	15	15	8.9	4.2	1.5	3.2	2.3	2.7	2.8
2000	18.4	15	15	10.9	3.6	2.9	1.9	1.8	3.0	3.4
2000	18.4	75	75	3.0	1.7	1.2	.6	.7	.6	0.0
2000	18.4	135	135	2.2	.6	1.3	.4	.1	0.0	1.0
2000	18.5	195	195	1.2	.8	1.0	.4	0.0	0.0	.6

DRI: Influent Density per 1 cc. of sample by direct reading Densiometer.

DRE: Effluent density per 1 cc. of sample by direct reading Densiometer

DEN: Density per 1 cc. of sample at entry of Ferrogram

D54: Density per 1 cc. of sample at 54 mm. on Ferrogram

D50: As D54 except at 50 mm.

D30: As D54 except at 30 mm.

D10: As D54 except at 10 mm.

SUMMARY OF PUMP BREAK-IN DATA AND ANALYSIS

Pump #: M   Pump Type: Gear

Pump Speed: 2000 RPM   Fluid Temp.: 65 °C

Inlet Press: 5 PSI

Outlet Press. (PSI)	Q (GPM)	Time At Pressure (Min)	Sample Time (Min)	DRI	DRE	DEN	D54	D50	D30	D10
500	51.2	15	15	4.6	3.5		3.4			
1000	50.3	15	15	4.7	4.4		2.3			
1500	48.9	15	15							
2000	47.9	15	15	4.6	2.9		2.5			
2000	47.5	75	75	3.9	1.5		3.3			
2000	48.7	135	135							
2000	48.8	195	195	4.6	4.1		1.4			

DRI: Influent Density per 1 cc. of sample by direct reading Densiometer.

DRE: Effluent density per 1 cc. of sample by direct reading Densiometer

DEN: Density per 1 cc. of sample at entry of Ferrogram

D54: Density per 1 cc. of sample at 54 mm. on Ferrogram

D50: As D54 except at 50 mm.

D30: As D54 except at 30 mm.

D10: As D54 except at 10 mm.

APPENDIX B

SUMMARY OF PUMP CONTAMINANT TESTS

SUMMARY OF PUMP TEST DATA AND ANALYSIS

Pump #: A
Rated Flow (Q_R): 38.4
Pump Speed: 2250 RPM
Inlet Press.: 5 PSI
Contaminant Concentration 300 mg/ℓ

Pump Type: Gear
GPM @ 3000 PSI
Fluid Temp.: 65 °C
Outlet Press.: 3000 PSI

Cont. Size (μM)	Q_i (GPM)	Q_F (GPM)	$\frac{Q_F}{Q_R}$	ΔQ (GPM)	Cont. Exp. Time (Min)	Sample Time (Min)	DRI	DRE	DEN	D54	D50	D30	D10
0-5	38.4	38.4	1.00	+0.0	10	10	52.7	15.9	5.87	7.23	4.3	1.73	1.77
0-10	38.4	38.4	1.00	0.0	10	10	13.4	5.1	2.90	2.07	1.67	0.70	1.97
0-20	38.4	38.4	1.00	0.0	12	10	20.7	12.2	15.6	8.10	7.97	1.73	5.90
0-30	38.4	36.1	0.95	-2.3	26	26	76.5	16.2	14.9	10.3	7.50	0.60	6.00
0-40	36.1	30.4	0.80	-5.7	24	24	84.4	27.8	22.8	13.6	15.8	3.27	7.50
0-50	30-4	28.1	0.74	-2.3	24	24	118.3	29.5	22.3	20.3	12.6	6.63	4.73
0-60	28.1	24.8	0.65	-3.3	28	28	128.8	39.5	28.1	22.9	18.4	13.0	7.67

DRI: Influent Density per 1 cc. of sample by direct reading Densiometer
DRE: Effluent density per 1 cc. of sample by direct reading Densiometer
DEN: Density per 1 cc. of sample at entry of Ferrogram
D54: Density per 1 cc. of sample at 54 mm. on Ferrogram
D50: As D54 except at 50 mm.
D30: As D54 except at 30 mm.
D10: As D54 except at 10 mm.
Q_i: Initial Flow
Q_F: Final Flow
Q_R: Rated Flow

SUMMARY OF PUMP TEST DATA AND ANALYSIS

Pump #: B
Rated Flow (Q_R): 18.0
Pump Speed: 3000 RPM
Inlet Press.: 5 PSI
Contaminant Concentration 300 mg/ℓ

Pump Type: Gear
GPM @ 3000 PSI
Fluid Temp.: 65 °C
Outlet Press.: 2000 PSI

Cont. Size (μM)	Q_i (GPM)	Q_F (GPM)	$\frac{Q_F}{Q_R}$	ΔQ (GPM)	Cont. Exp. Time (Min)	Sample Time (Min)	DRI	DRE	DEN	D54	D50	D30	D10
0-5	18.0	18.0	1.00	0.0	38	38	30.0	10.7	5.67	4.63	3.1	6.0	3.5
0-10	18.0	18.0	1.00	0.0	26								
0-20	18.0	18.0	1.00	0.0	16	16	94.6	19.7	21.3	17.5	12.1	5.27	1.3
0-30	18.0	16.9	.94	-1.1	34	34	132.3	27.0	25.0	18.8	13.6	4.13	8.63
0-40	16.9	15.7	.87	-1.2	28	28	134.0	26.0	25.4	17.3	11.1	4.27	4.87
0-50	15.7	15.4	.86	-0.3	20	20	45.7	13.5	10.7	10.9	5.0	5.23	3.07
0-60	15.4	13.5	.75	-1.9	20	20	36.5	35.2	20.5	11.7	8.0	3.63	3.20
0-70	13.5	12.7	.71	-0.8	16	16	31.6	14.3	14.5	7.0	6.27	1.27	0.00

DRI: Influent Density per 1 cc. of sample by direct reading Densiometer
DRE: Effluent density per 1 cc. of sample by direct reading Densiometer
DEN: Density per 1 cc. of sample at entry of Ferrogram
D54: Density per 1 cc. of sample at 54 mm. on Ferrogram
D50: As D54 except at 50 mm.
D30: As D54 except at 30 mm.
D10: As D54 except at 10 mm.
Q_i: Initial Flow
Q_F: Final Flow
Q_R: Rated Flow

SUMMARY OF PUMP TEST DATA AND ANALYSIS

Pump #: C

Rated Flow (Q_R): 11.6

Pump Speed: 2000 RPM

Inlet Press.: 5 PSI

Contaminant Concentration 300 mg/ℓ

Pump Type: Gear

GPM @ 2000 PSI

Fluid Temp.: 65 °C

Outlet Press.: 2000 PSI

Cont. Size (μM)	Q_i (GPM)	Q_F (GPM)	$\frac{Q_F}{Q_R}$	ΔQ (GPM)	Cont. Exp. Time (Min)	Sample Time (Min)	DRI	DRE	DEN	D54	D50	D30	D10
0-5	11.6	11.6	1.00	+0.0	30	30	13.6	6.9	2.77	2.40	1.97	1.90	4.13
0-10	11.6	11.5	1.00	-0.1	12								
0-20	11.5	11.6	1.00	+0.1	22	22	97.2	18.4	18.5	18.5	10.1	5.53	5.60
0-30	11.6	11.4	1.00	-0.2	26	26	132.4	23.9	23.5	18.1	12.5	6.57	4.23
0-40	11.4	10.7	.94	-0.7	20	20	144.5	27.1	26.3	20.0	12.3	3.47	8.10
0-50	10.7	9.9	.87	-0.8	16	16	127.2	22.2	22.9	20.3	11.4	3.0	7.03
0-60	9.9	8.0	0.70	-1.9	16	16	118.2	17.8	23.0	17.2	9.6	5.8	4.43

DRI: Influent Density per 1 cc. of sample by direct reading Densiometer

DRE: Effluent density per 1 cc. of sample by direct reading Densiometer

DEN: Density per 1 cc. of sample at entry of Ferrogram

D54: Density per 1 cc. of sample at 54 mm. on Ferrogram

D50: As D54 except at 50 mm.

D30: As D54 except at 30 mm.

D10: As D54 except at 10 mm.

Q_i: Initial Flow

Q_F: Final Flow

Q_R: Rated Flow

SUMMARY OF PUMP TEST DATA AND ANALYSIS

Pump #: D
Rated Flow (Q_R): 18.3
Pump Speed: 2000 RPM
Inlet Press.: 5 PSI
Contaminant Concentration 300 mg/ℓ

Pump Type: Gear
GPM @ 2000 PSI
Fluid Temp.: 65 °C
Outlet Press.: 2000 PSI

Cont. Size (μM)	Q_i (GPM)	Q_F (GPM)	$\frac{Q_F}{Q_R}$	ΔQ (GPM)	Cont. Exp. Time (Min)	Sample Time (Min)	DRI	DRE	DEN	D54	D50	D30	D10
0-5	18.3	18.3	1.0	0.0	16	16	64.8	16.2	8.33	7.87	4.5	0.0	1.83
0-10	18.3	18.3	1.0	0.0	16	16	43.7	17.1	11.0	10.6	6.2	8.07	4.3
0-20	18.3	18.3	1.0	0.0	16	16	113.0	23.0	24.3	15.1	13.3	4.77	11.6
0-30	18.3	18.2	0.99	-0.1	16	16	139.0	28.8	31.9	28.4	11.3	12.9	6.93
0-40	18.2	17.6	0.96	-0.6	32	32	149.4	34.9	34.4	33.7	26.0	13.1	14.6
0-50	17.6	16.6	0.91	-1.0	32	32	146.0	28.1	35.0	33.1	20.0	0.0	5.2
0-60	16.6	15.5	0.85	-1.1	22	22	90.3	23.9	27.7	26.5	26.5	7.60	14.7
0-70	15.5	14.4	0.79	-1.1	i2	22	123.9	26.0	35.67	24.1	16.9	12.8	4.53

DRI: Influent Density per 1 cc. of sample by direct reading Densiometer
DRE: Effluent density per 1 cc. of sample by direct reading Densiometer
DEN: Density per 1 cc. of sample at entry of Ferrogram
D54: Density per 1 cc. of sample at 54 mm. on Ferrogram
D50: As D54 except at 50 mm.
D30: As D54 except at 30 mm.
D10: As D54 except at 10 mm.
Q_i: Initial Flow
Q_F: Final Flow
Q_R: Rated Flow

SUMMARY OF PUMP TEST DATA AND ANALYSIS

Pump #: E
Rated Flow (Q_R): 19.2
Pump Speed: 2000 RPM
Inlet Press.: 5 PSI
Contaminant Concentration 300 mg/ℓ

Pump Type: Gear
GPM @ 2000 PSI
Fluid Temp.: 65 °C
Outlet Press.: 2000 PSI

Cont. Size (μM)	Q_i (GPM)	Q_F (GPM)	$\frac{Q_F}{Q_R}$	ΔQ (GPM)	Cont. Exp. Time (Min)	Sample Time (Min)	DRI	DRE	DEN	D54	D50	D30	D10
0-5	19.2	19.2	1.0	0.0	16	16	28.7	9.3	8.33	6.77	5.07	1.83	9.27
0-10	19.2	19.0	0.99	-0.2	10								
0-20	19.0	18.1	0.94	-0.9	30	30	23.8	5.0	15.3	11.6	11.3	11.7	5.63
0-30	18.1	16.0	0.83	-2.1	30	30	261.0	40.0	71.0	37.3	25.0	9.67	9.67
0-40	16.0	13.7	0.71	-2.3	22	22	680.0	410.0	65.7	9.33	1.33	5.67	9.0

DRI: Influent Density per 1 cc. of sample by direct reading Densiometer
DRE: Effluent density per 1 cc. of sample by direct reading Densiometer
DEN: Density per 1 cc. of sample at entry of Ferrogram
D54: Density per 1 cc. of sample at 54 mm. on Ferrogram
D50: As D54 except at 50 mm.
D30: As D54 except at 30 mm.
D10: As D54 except at 10 mm.
Q_i: Initial Flow
Q_F: Final Flow
Q_R: Rated Flow

SUMMARY OF PUMP TEST DATA AND ANALYSIS

Pump #: G
Rated Flow (Q_R): 23
Pump Speed: 2000 RPM
Inlet Press.: 5 PSI
Contaminant Concentration 300 mg/ℓ

Pump Type: Gear
GPM @ 2000 PSI
Fluid Temp.: 65 °C
Outlet Press.: 2000 PSI

Cont. Size (μM)	Q_i (GPM)	Q_F (GPM)	$\frac{Q_F}{Q_R}$	ΔQ (GPM)	Cont. Exp. Time (Min)	Sample Time (Min)	DRI	DRE	DEN	D54	D50	D30	D10
0-5	23.0	23.0	1.0	0.0	12	12	26.8	7.2	5.53	5.57	3.93	0.57	2.77
0-10	23.0	23.0	1.0	0.0	16	16	16.1	9.0	3.07	3.47	3.13	1.73	2.4
0-20	23.0	23.0	1.0	0.0	14	14	26.6	12.4	1.5	0.9	0.47	0.0	0.43
0-30	23.0	22.5	.98	-0.5	20	20	51.5	6.1	13.3	12.2	8.73	3.17	6.73
0-40	22.5	21.5	.93	-1.0	30								
0-50	21.5	20.5	.89	-1.0	30	30	56.4	18.0	14.4	14.9	11.5	6.23	1.17
0-60	20.5	19.5	.85	-1.0	20	20	50.2	20.3	16.8	17.8	10.8	5.83	5.17
0-70	19.5	18.0	.78	-1.5	30	30	50.7	12.8	9.87	14.9	8.17	0.0	0.0
0-80	18.0	17.0	.74	-1.0	16	16	46.7	13.3	16.0	12.8	7.87	0.73	6.17

DRI: Influent Density per 1 cc. of sample by direct reading Densiometer
DRE: Effluent density per 1 cc. of sample by direct reading Densiometer
DEN: Density per 1 cc. of sample at entry of Ferrogram
D54: Density per 1 cc. of sample at 54 mm. on Ferrogram
D50: As D54 except at 50 mm.
D30: As D54 except at 30 mm.
D10: As D54 except at 10 mm.
Q_i: Initial Flow
Q_F: Final Flow
Q_R: Rated Flow

SUMMARY OF PUMP TEST DATA AND ANALYSIS

Pump #: H
Rated Flow (Q_R): 40.8
Pump Speed: 1800 RPM
Inlet Press.: 5 PSI
Contaminant Concentration 300 mg/ℓ

Pump Type: Gear
GPM @ 2000 PSI
Fluid Temp.: 65 °C
Outlet Press.: 2000 PSI

Cont. Size (μM)	Q_i (GPM)	Q_F (GPM)	$\frac{Q_F}{Q_R}$	ΔQ (GPM)	Cont. Exp. Time (Min)	Sample Time (Min)	DRI	DRE	DEN	D54	D50	D30	D10
0-5	40.8	40.8	1.00	0.0	14	14	38.5	11.4	9.6	7.1	4.6	8.2	4.2
0-10	40.8	40.8	1.00	0.0	18	18	48.6	17.2	15.0	12.2	7.6	.6	10.9
0-20	40.8	40.5	.99	-0.3	12	12	76.7	20.0	13.2	4.1	6.0	4.8	1.2
0-30	40.5	40.3	.99	-0.2	22	22	62.4	15.6	20.0	11.5	11.2	7.9	4.0
0-40	40.3	38.8	.95	-1.5	24	24	33.8	8.4	17.5	9.4	7.5	1.8	1.4
0-50	38.8	38.4	.94	-0.4	14	14	61.1	14.8	16.7	11.3	8.4	5.2	3.4
0-60	38.4	36.3	.89	-2.1	30	30	49.2	14.5	13.0	11.1	8.3	7.4	6.1
0-70	36.3	36.1	.88	-0.2	12	12	28.0	8.5	13.6	8.5	7.4	4.3	5.8
0-80	36.1	34.8	.85	-1.3	22	22	41.3	15.0	20.8	13.8	10.8	6.8	5.5

DRI: Influent Density per 1 cc. of sample by direct reading Densiometer
DRE: Effluent density per 1 cc. of sample by direct reading Densiometer
DEN: Density per 1 cc. of sample at entry of Ferrogram
D54: Density per 1 cc. of sample at 54 mm. on Ferrogram
D50: As D54 except at 50 mm.
D30: As D54 except at 30 mm.
D10: As D54 except at 10 mm.
Q_i: Initial Flow
Q_F: Final Flow
Q_R: Rated Flow

SUMMARY OF PUMP TEST DATA AND ANALYSIS

Pump #: J
Rated Flow (Q_R): 26.5
Pump Speed: 2000 RPM
Inlet Press.: 5 PSI
Contaminant Concentration 300 mg/ℓ

Pump Type: Gear
GPM @ 2000 PSI
Fluid Temp.: 65 °C
Outlet Press.: 2000 PSI

Cont. Size (μM)	Q_i (GPM)	Q_F (GPM)	$\frac{Q_F}{Q_R}$	ΔQ (GPM)	Cont. Exp. Time (Min)	Sample Time (Min)	DRI	DRE	DEN	D54	D50	D30	D10
0-5	26.5	26.1	.98	-0.4	14	14	22.5	8.3	6.3	5.9	5.8	4.1	2.9
0-10	26.1	26.0	.98	-0.1	14	14	27.6	8.8	12.2	11.2	7.4	7.3	3.6
0-20	26.0	25.6	.97	-0.4	24	24	51.1	12.5	14.7	15.2	9.2	6.2	3.1
0-30	25.6	23.5	.89	-2.1	24	24	118.0	34.0	89.0	27.3	20.0	6.7	0.0
0-40	23.5	21.0	.79	-2.5	16	16	136.0	51.0	85.3	35.3	30.7	28.3	17.3
0-50	21.0	18.5	.70	-2.5	16	16	76.0	35.0	29.7	7.0	10.0	18.3	7.0

DRI: Influent Density per 1 cc. of sample by direct reading Densiometer
DRE: Effluent density per 1 cc. of sample by direct reading Densiometer
DEN: Density per 1 cc. of sample at entry of Ferrogram
D54: Density per 1 cc. of sample at 54 mm. on Ferrogram
D50: As D54 except at 50 mm.
D30: As D54 except at 30 mm.
D10: As D54 except at 10 mm.
Q_i: Initial Flow
Q_F: Final Flow
Q_R: Rated Flow

SUMMARY OF PUMP TEST DATA AND ANALYSIS

Pump #: L
Rated Flow (Q_R): 18.5
Pump Speed: 2000 RPM
Inlet Press.: 5 PSI
Contaminant Concentration 75 mg/ℓ

Pump Type: Gear
GPM @ 2000 PSI
Fluid Temp.: 65 °C
Outlet Press.: 2000 PSI

Cont. Size (μM)	Q_i (GPM)	Q_F (GPM)	$\frac{Q_F}{Q_R}$	ΔQ (GPM)	Cont. Exp. Time (Min)	Sample Time (Min)	DRI	DRE	DEN	D54	D50	D30	D10
0-5	18.5	18.5	1.0	0.0	30	28	12.4	4.5	4.9	2.0	2.4	2.6	5.5
0-10	18.5	18.5	1.0	0.0	30	28	58.1	11.9	8.9	6.3	5.8	3.5	2.9
0-20	18.5	18.5	1.0	0.0	30	28	25.2	7.1	5.1	5.8	3.7	5.9	2.8
0-30	18.5	18.5	1.0	0.0	30	28	19.7	9.6	5.6	4.2	4.2	4.2	3.1
0-40	18.5	18.5	1.0	0.0	30	28	10.2	8.1	3.2	2.8	2.1	3.2	1.9
0-50	18.5	18.4	.99	-.1	30	28	8.1	4.6	1.9	2.2	3.4	3.0	3.6
0-60	18.4	18.3	.99	-.1	30	28	47.2	10.8	8.21	8.5	5.7	4.3	3.4
0-70	18.3	18.0	.97	-.3	30	28	13.6	4.8	29.3	38.3	19.0	6.7	10.3
0-80	18.0	17.8	.96	-.2	30	28	43.9	13.6	9.6	8.3	6.2	3.6	2.9

DRI: Influent Density per 1 cc. of sample by direct reading Densiometer
DRE: Effluent density per 1 cc. of sample by direct reading Densiometer
DEN: Density per 1 cc. of sample at entry of Ferrogram
D54: Density per 1 cc. of sample at 54 mm. on Ferrogram
D50: As D54 except at 50 mm.
D30: As D54 except at 30 mm.
D10: As D54 except at 10 mm.
Q_i: Initial Flow
Q_F: Final Flow
Q_R: Rated Flow

SUMMARY OF PUMP TEST DATA AND ANALYSIS

Pump #: M
Rated Flow (Q_R): 48.4
Pump Speed: 2000 RPM
Inlet Press.: 5 PSI
Contaminant Concentration 75 mg/ℓ

Pump Type: Gear
GPM @ 2000 PSI
Fluid Temp.: 65 °C
Outlet Press.: 2000 PSI

Cont. Size (μM)	Q_i (GPM)	Q_F (GPM)	$\frac{Q_F}{Q_R}$	ΔQ (GPM)	Cont. Exp. Time (Min)	Sample Time (Min)	DRI	DRE	DEN	D54	D50	D30	D10
0-5	48.4	48.4	1.0	+ .0	30	28	13.3	6.8	5.1	1.7	1.6	2.1	1.9
0-10	48.4	48.4	1.0	0.0	30	28	23.7	5.3	2.8	2.3	1.3	1.9	1.3
0-20	48.4	47.9	.99	- .5	30	28	14.1	7.0	5.8	4.8	3.3	5.1	3.3
0-30	47.9	46.4	.96	-1.5	30	28	36.9	8.5	8.5	5.1	4.5	3.6	3.9
0-40	46.4	44.7	.93	-1.7	30	28	16.8	5.4	6.4	4.5	2.2	0.7	2.1
0-50	44.7	41.8	.87	-2.9	30	28	47.7	8.0	10.6	9.7	5.6	3.0	2.6
0-60	41.8	39.7	.82	-2.1	30	28	36.0	8.2	10.9	9.1	5.8	7.6	3.6
0-70	39.7	37.7	.78	-2.0	30	28	18.4	5.3	4.8	3.6	2.8	3.2	3.2

DRI: Influent Density per 1 cc. of sample by direct reading Densiometer
DRE: Effluent density per 1 cc. of sample by direct reading Densiometer
DEN: Density per 1 cc. of sample at entry of Ferrogram
D54: Density per 1 cc. of sample at 54 mm. on Ferrogram
D50: As D54 except at 50 mm.
D30: As D54 except at 30 mm.
D10: As D54 except at 10 mm.
Q_i: Initial Flow
Q_F: Final Flow
Q_R: Rated Flow

SECTION IV

LUBE OIL FILTER STUDY FOR MOBILE ON-OFF HIGHWAY DIESEL ENGINE DRIVEN VEHICLES – II

PROJECT STAFF

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FOREWORD

This report presents the results of a study to develop a complete set of testing procedures and specifications for lube oil filters which would be industrially acceptable and compatible with U.S. Army MERDC equipment. The effort has been directed towards lube oil filters utilized for on-off highway diesel engine driven vehicles. The primary emphasis of this year's effort has been in the verification of the test procedures and promulgation of the results on an industrial basis to gain the support and understanding of industry.

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CHAPTER I

INTRODUCTION

The overall objective of the 1975-76 MERDC-OSU Lube Oil Filter Program was to complete the formalization of a set of testing procedures and specifications for lube oil filters which would be industrially acceptable and compatible with U.S. Army requirements. This project was initiated during the 1973-74 MERDC-OSU Program, and several basic test procedures were developed and a number of verification tests conducted. The current phase represents an extension of that work with emphasis upon the verification of the critical test methods and the promotion of the methods on an industrial level.

The general approach used for the development of the lube oil filter test documents relied heavily upon existing test methods for both lubricating and hydraulic oil filters as well as current industrial opinions. After an extensive literature survey was conducted, a questionnaire was mailed to a broad range of industrial representatives. First drafts of the necessary test methods were formulated and meetings of the industrial advisors held to review and revise the documents. Extensive testing at OSU was initiated, and the procedures were again revised. The results of this effort are presented in the last year's MERDC-OSU report, Ref. [1].

During the past year, the project was continued and additional procedure verification tests conducted to establish both the repeatability (within one laboratory) and reproducibility (between laboratories) of the critical methods. In addition, a technical paper was written and presented to the Society of Automotive Engineers [2] in order to enhance industrial understanding and acceptance. MERDC-OSU personnel also were active on the SAE lube oil filter test methods subcommittee to whom the procedures have been submitted for adoption. Finally, an engine wear survey was conducted of industrial representatives, and the resulting data

were utilized in an attempt to establish proposed filtration requirements for MERDC equipment.

Chapter II of this report phase outlines the test procedures recommended, and a complete copy of the revised multi-pass method is given in Appendix B. Chapter III discusses the results of the verification testing, and the actual data are included in Appendix A. The results of an attempt to establish MERDC engine filtration requirements based on available engine wear data are presented in Chapter IV, and a proposed specification is given in Appendix C. Finally, conclusions and recommendations are given in Chapter V.

CHAPTER II

LUBE OIL FILTER TESTING PROCEDURES

A set of 11 test procedures was proposed in Ref. [1] for evaluating the performance of lube oil filters. These procedures are the following:

1. **Multi-Pass Method for Evaluating the Particle Separation Characteristics of a Lube Oil Filter Element**

2. **A Recirculating Method for Evaluating the Sludge Removal Characteristics of a Lube Oil Filter Element**

3. **Method for Determining the Fabrication Integrity of a Lube Oil Filter Element**

4. **Method for Verifying the Collapse/Burst Resistance of a Lube Oil Filter Element**

5. **Method for Verifying the Material Compatibility of a Lube Oil Filter Element**

6. **Method for Determining the Media Migration of a Lube Oil Filter Element**

7. **Method for Determining the Ash Type Oil Additive Removal Characteristics of a Lube Oil Filter Element**

8. **Method for Determining the Performance of Anti-Drainback Valves on Lube Oil Filters**

9. **Method for Verifying the Vibration Fatigue Resistance of a Lube Oil Filter**

10. **Method for Verifying the Hydrostatic Burst Resistance of a Lube Oil Filter**

11. **Method for Evaluating the Performance Characteristics of a Lube Oil Filter Relief Valve**

The effort of the MERDC-OSU Program during the past two years has been primarily concentrated on the development and verification of the first procedure. The second method is currently being considered by the SAE lube oil filter test methods subcommittee, and that group is attempting to develop a workable procedure. The remaining nine proposed procedures are similar to existing SAE standard methods; therefore, no verification work was conducted by OSU. The most recent version of the multi-pass test method is contained in Appendix B of this report, while brief descriptions of the other ten procedures can be found in Ref. [1].

The multi-pass test for evaluating the particle separation characteristics of a lube oil filter is believed to be the most important of all the tests. Unless a candidate filter is capable of controlling the abrasive particulate contamination level (which is its primary function), its performance on the remaining tests is of little consequence. The test developed and presented in Appendix B basically involves the subjection of the test filter to constant rated flow and continuous contaminant injection. The contaminant not removed by the filter is allowed to recirculate back to the reservoir where new contaminant is added – thus the name multi-pass. Samples are extracted from upstream and downstream of the test filter, and particle counts are performed on the samples. The results are reported in terms of filtration or alpha (α) ratio, which is the ratio of the cumulative upstream count to the corresponding downstream count greater than a given particle size. Alpha (α) is utilized to represent the filtration ratio for lube oil filters tested with AC Coarse Test Dust to distinguish it from the Beta term utilized for hydraulic filters with AC Fine Test Dust.

The multi-pass test method has been proposed to SAE, and the appropriate subcommittee is now considering adoption of the procedure as a standard. The procedure as it was initially

proposed called for a test system fluid volume equal to one-fourth the rated flow (per minute) value. It became obvious that, for low-rate filters such as those used on small diesel engines or automobiles, the required volume would be less than practical for most test facilities. A revision was suggested at SAE to specify the system volume as one-fourth the flow rate plus three litres; thus, there would never be less than three litres of fluid in the test system. Since the volume of the test system is actually critical only during the very early stabilization part of a normal test, this revision should create no repeatability problems while greatly extending the applicability of the test.

In order to further promote the multi-pass test method for lube oil filters on an industrial level, a technical paper was written by OSU personnel. This paper, entitled *"Lube Oil Filter Evaluation"* [2], was presented to the 1975 SAE Off-Highway Vehicle Meeting in Milwaukee, Wisconsin, September 8-11, 1975. This paper presented the results of the initial MERDC-OSU study in order to familiarize the industry with the test methods. Preliminary repeatability and discriminatory characteristics of the test method were also included. The response to the paper presentation was very encouraging, and it is believed that the industrial understanding and acceptance of the test procedure was enhanced by the presentation.

CHAPTER III

MULTI-PASS TEST PROCEDURE VERIFICATION

A number of multi-pass tests were conducted during the initial year of the MERDC-OSU Lube Oil Filter Project. The results of these tests which established the initial repeatability and discriminatory characteristics of the procedure are presented in Ref. [1]. During the immediate past year, several additional tests were conducted with emphasis on completing the previous effort and in addition to establish the reproducibility characteristics of the procedure between various laboratories.

DISCRIMINATION

Table 3-1 is a compilation of the results from all the filter tests which have been conducted at OSU in accordance with the latest multi-pass test. Complete data sheets are included in Appendix A. A quick scan through these results will reveal the wide range of filters which are available on today's market. All of these filters were submitted to OSU by industrial representatives of both the filter manufacturer and end-item advisory groups and were supposed to be representative of typical lube oil filters for on-off highway diesel engine driven vehicles. The filters reported in Table 3-1 were all tested at the specified rated flow and a terminal pressure drop of 2.76 bar differential (40 psid). The relief valve was blocked, except in those cases where the valve was an integral part of a spin-on can type filter. In these cases, the terminal pressure drop was generally reduced to below the rated relief valve cracking pressure.

TABLE 3-1. COMPILATION OF OSU MULTI-PASS TEST RESULTS ON LUBE OIL FILTERS.

FPRC NO.	RATED FLOW (ℓpm)	ARITHMETIC AVERAGED FILTRATION RATIOS					ACCTD CAPACITY (grams)
		α_{10}	α_{20}	α_{30}	α_{40}	α_{50}	
328C	26.5	8.62	44.10	183.00	303.00	434.00	57.2
328G	26.5	5.72	24.40	96.40	173.00	293.00	50.0
328I	26.5	5.62	27.50	112.00	195.00	177.00	46.3
328J	26.5	7.59	31.00	141.00	363.00	387.00	55.7
329B	26.5	8.34	35.80	126.00	192.00	181.00	65.0
330B	56.8	1.24	1.91	6.37	—	—	162.9
331A	37.9	1.28	1.76	3.65	13.00	37.60	62.0
332A	37.9	1.40	2.91	15.00	98.80	272.00	95.0
333A	75.7	1.58	6.10	41.40	118.50	—	64.6
333B	75.7	1.62	5.97	33.20	95.40	—	60.2
333C	75.7	1.52	8.45	41.80	94.90	—	60.7
334C	75.7	2.00	6.00	29.70	112.00	288.00	86.8
335A	37.9	23.20	20.70	21.60	19.90	18.30	18.4
336A	37.9	16.40	15.80	15.50	16.40	—	29.4
337B	45.4	22.20	581.00	∞	∞	—	84.0
338A	45.4	2.48	84.30	510.00	699.00	∞	37.6
339A	75.7	8.85	26.60	59.30	55.60	45.30	230.0
340F	56.8	1.22	2.02	5.29	13.80	22.70	123.8
341A	56.8	1.99	14.00	∞	∞	—	89.6
342B	106.0	36.50	107.00	350.00	899.00	∞	507.8
343D	106.0	2.80	5.87	13.50	46.50	127.00	138.5
344B	90.9	—	—	1.80	5.06	11.50	120.5
345A	90.9	3.44	10.80	34.60	53.00	55.90	158.5
346A	43.5	1.22	2.17	5.01	14.80	∞	49.5
347A	43.5	1.12	1.51	2.59	5.82	10.40	38.9
350B	94.6	5.15	18.20	39.00	59.60	—	206.0
351A	94.6	1.86	1.72	1.80	1.46	1.35	336.0
352A	94.6	1.32	1.54	1.77	2.09	2.15	1244.0
384B	151.4	1.90	72.60	134.00	148.00	111.00	300.7
384C	151.4	2.68	64.00	81.60	80.50	70.00	283.3
385A	151.4	1.21	1.80	5.28	27.80	78.90	299.6
386A	151.4	20.00	154.00	350.00	306.00	333.00	175.3+
401A	18.9	1.69	6.58	51.10	101.00	∞	37.2
402C	18.9	1.32	4.11	18.80	43.50	140.00	23.9
403E	18.9	1.55	2.21	2.97	4.22	6.31	11.4
404A	18.9	1.58	3.16	12.72	60.78	∞	70.4
405D	18.9	1.47	2.17	5.07	24.50	∞	46.8
406E	18.9	2.55	4.70	19.80	47.50	132.00	67.0
407E	18.9	2.39	2.75	4.80	7.99	9.98	69.6
408E	18.9	2.47	5.14	9.63	44.60	∞	80.3
410A	45.4	1.27	1.48	2.16	3.19	5.50	135.0
410B	45.4	1.20	1.35	1.79	2.96	4.71	118.8
410C	45.4	1.25	1.38	1.78	3.09	7.31	121.3

From the results shown in Table 3-1, it can be seen that the lowest average α_{10} value measured was 1.12. Since the filtration ratio is the ratio of the cumulative upstream 10μM count to the respective downstream count, the possibly more familiar term of separation efficiency can be calculated by $[(\alpha - 1)/\alpha] \times [100\%]$. Thus, the α_{10} value of 1.12 corresponds to a cumulative 10μM efficiency of only 10.71%. The maximum α_{10} value encountered was 36.5, which is equal to an efficiency of 97.26%. This certainly represents a wide spread in filter performance.

Likewise, the ACCTD capacity, which is the total quantity of AC Coarse Test Dust added to the system when the terminal pressure drop is attained, shows a wide range for the filters tested. The minimum value obtained was 11.4 grams and is more than two orders of magnitude lower than the maximum of 1244 grams. Of course, the filter size and separation performance must be considered when comparing capacities; however, this is still quite a significant difference.

In order to graphically illustrate the wide range of filters available and to demonstrate the capability of the multi-pass procedure to discriminate between different filters, a plot of average filtration ratio versus particle size for various filters is given in Fig. 3-1. These filters represent just a few of those from Table 3-1 but cover a wide cross-section of performance capabilities. It can be easily seen, from either Table 3-1 or Fig. 3-1, that the multi-pass test method has good discrimination characteristics.

REPEATABILITY

It is a necessary but not sufficient condition for a test method to discriminate between unlike test specimens. In addition, and of equal importance, the procedure must provide similar results for identical specimens. Several tests were performed to demonstrate the repeatability characteristics of the multi-pass test. Multiple tests were conducted on element numbers 328, 333, 384, and 410. The results can be found in Table 3-1, and the α ratio

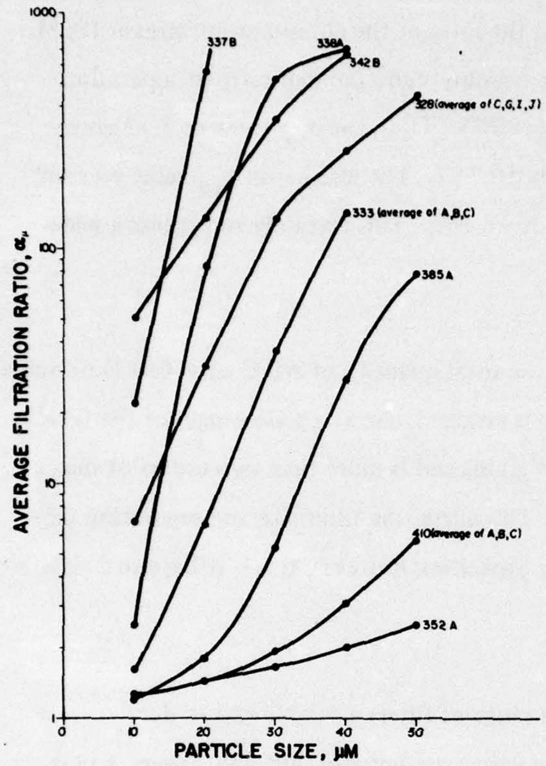

Fig. 3-1. Discriminatory Characteristics of Multi-Pass Test Method.

plots are given in Figs. 3-2 through 3-4. These data undeniably reveal the excellent repeatability characteristics of the test method. It can be seen from the results of all the filters evaluated for repeatability characteristics that the deviation or spread in the data is more appreciable at the larger particle sizes and larger values of alpha. This data scatter is primarily due to the low number of particles in the downstream samples which occurs when the alpha values are high. Sampling and statistical counting errors are more significant when this condition occurs. It is not unusual for the alpha curves to *"break over"* at high values due to such imperfections in fluid sampling and analysis. This phenomenon may also be caused by leakage through and around element seals or through leaky relief valves.

REPRODUCIBILITY

Another primary characteristic of a candidate test method is that it must be capable of resulting in similar data when identical specimens are evaluated in different laboratories. Data were obtained from other laboratories on two of the filters which OSU had tested a number of

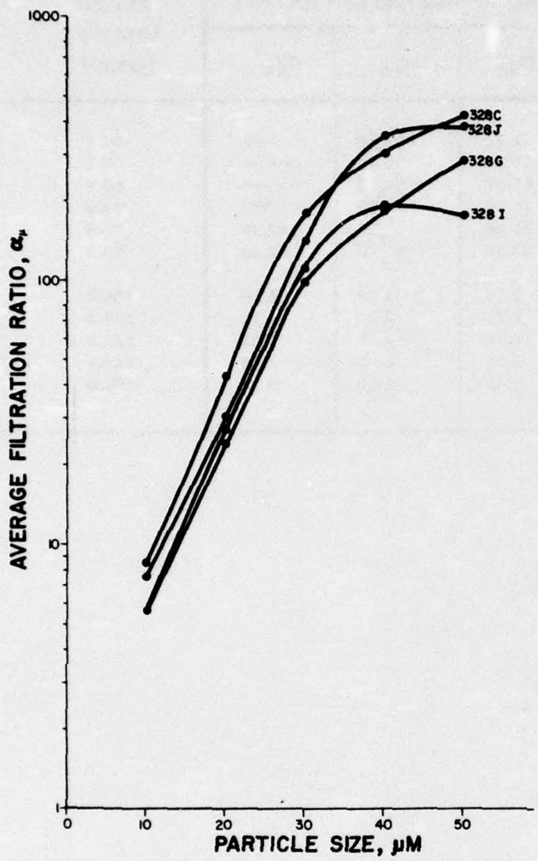

Fig. 3-2. Repeatability Characteristics for Element No. 328.

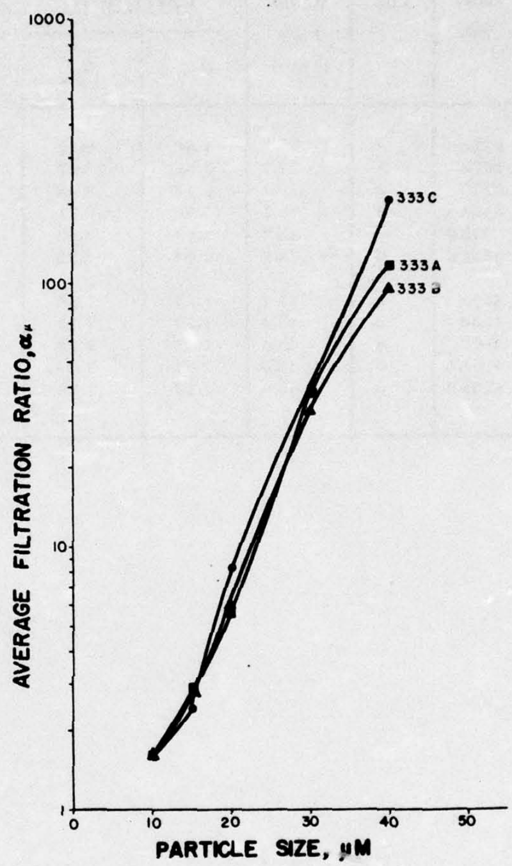

Fig. 3-3. Repeatability Characteristics for Element No. 333.

times for repeatability. These results are listed in Table 3-2 for element numbers 333 and 410. Element 333 was evaluated in a total of three laboratories including OSU. The results delineated in Table 3-1 are plotted in Fig. 3-5 and demonstrate excellent reproducibility between the different laboratories. Element 410, which was much *"coarser"* than number 333, was tested in two laboratories. The resulting data are also listed in Table 3-2 and are plotted in Fig. 3-6. When these data are compared to the total data summary and the discriminatory characteristics in Fig. 3-1, the excellent reproducibility cannot be questioned.

TABLE 3-2. REPRODUCIBILITY DATA FROM MULTI-PASS TEST.

Filter No.	Lab	Rated Flow (ℓpm)	ARITHMETIC AVERAGED FILTRATION RATIOS					ACCTD Capacity (grams)
			α_{10}	α_{20}	α_{30}	α_{40}	α_{50}	
333A	A	75.7	1.58	6.10	41.40	118.50	——	64.6
333B	A	75.7	1.62	5.97	33.20	95.40	——	60.2
333C	A	75.7	1.52	8.45	41.80	94.90	——	60.7
333AA	B	75.7	2.65	9.01	46.70	55.30	∞	74.9
333BB	C	75.7	1.59	5.86	25.40	80.10	112.00	70.6
333CC	C	75.7	1.64	6.22	32.90	91.40	82.30	70.2
410A	A	45.4	1.27	1.48	2.16	3.19	5.50	135.0
410B	A	45.4	1.20	1.35	1.79	2.96	4.71	118.8
410C	A	45.4	1.25	1.38	1.78	3.09	7.31	121.3
410AA	C	45.4	1.47	1.67	2.15	2.95	4.03	115.4
410BB	C	45.4	1.10	1.20	1.40	2.05	2.25	121.0

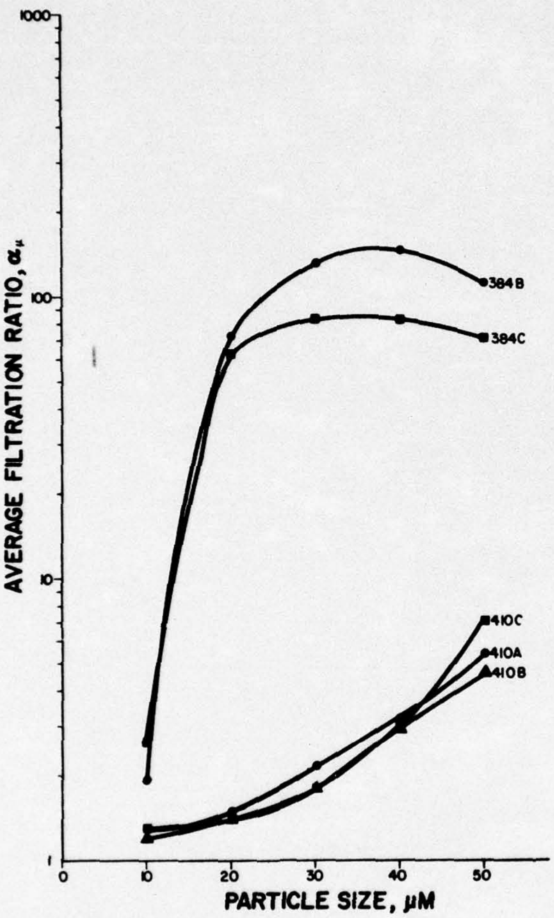

Fig. 3-4. Repeatability Characteristics for Elements Nos. 384 and 410.

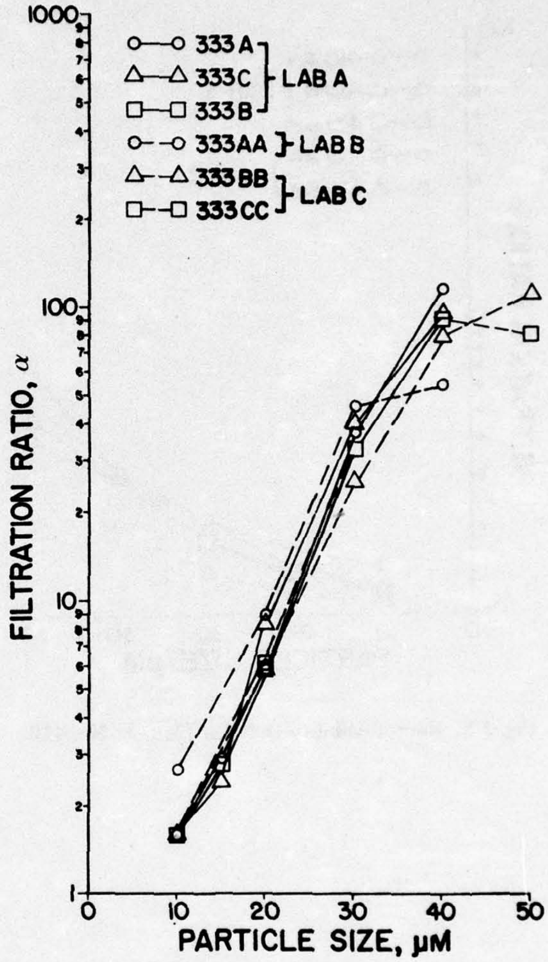

Fig. 3-5. Reproducibility Data for Element No. 333.

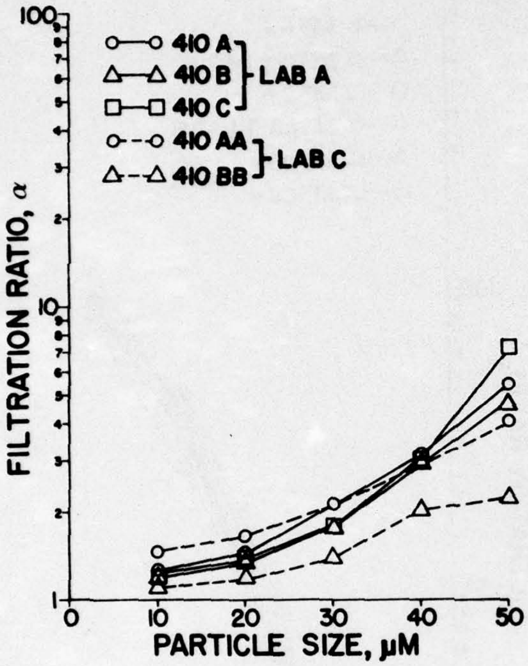

Fig. 3-6. Reproducibility Data for Element No. 410.

CHAPTER IV

ESTABLISHMENT OF ENGINE FILTRATION REQUIREMENTS FOR MERDC EQUIPMENT

One additional study to be conducted during this phase of the MERDC-OSU Program was an attempt to establish filtration requirements for typical on-off highway diesel engines utilizing any available data. Basically, in order to determine the filtration level required, several important pieces of information must be known. These include the contaminant ingression rate, the particle size distribution of the ingressed contaminant, and the contaminant sensitivity of the critical engine components.

The plan of attack taken by OSU personnel to establish the above requirements was first to conduct an extensive literature survey, then to send a questionnaire to all industrial participants. The literature survey was conducted and the general bibliography of this report delineates the most pertinent publications found. A letter was sent to all industrial advisors asking for engine wear data of which they were aware and which might be applicable to the MERDC-OSU effort. Very few responses were obtained which seem to indicate that little of these type studies have been conducted. The following paragraphs summarize the findings of these surveys.

The total contaminant ingression required includes that contaminant which enters the lubrication oil from the external environment, the contaminant which is generated internally from wearing surfaces, and the contaminant which is built in or is added with oil replenishment. There have been studies conducted and reported relative to typical ingression rates experienced in diesel engines; however, these studies generally consider only the contaminant which enters the system from the external environment either through air breathers or other

sources. Grim [3] reported that, in laboratory testing, one gram of dust injected per hour produced a wear rate comparable to the highest wear rate experienced by operators in areas where abrasive dust conditions are severe, engine operating conditions are bad, and maintenance is poor. In his tests, he was introducing standard AC Fine Test Dust as the abrasive contaminant. Buckman and Kemp [4] reported that sludge generation for 2.1 litre diesel engines was in the order of 3-4 grams per 100 miles. These engines though were generally on highway vehicles and cannot be directly compared with the tractor ingression rates reported by Grim. There have been a few other attempts to estimate the ingression into engines; however, most work has been conducted on gasoline engines or diesel engines utilized strictly for on-highway applications.

Very little literature could be found relative to the particle size distribution of ingressed contaminant for typical diesel engines. Of course, the distribution of contaminant entering through the air cleaner is directly dependent upon the contaminant exposed to the cleaner and the particle separation characteristics of the cleaner. Decker and Bailey [5] gave a table for the relative particle size distribution of dust samples at various heights above the ground level which would be a direct indication of the contaminant exposed to the air cleaner inlet. However, these distributions are for given conditions which may or may not be representative of typical MERDC equipment environments. Den Besten et al [6] present typical particle size distribution in the effluent of various air cleaners with AC Test Dust exposed to the inlet. These data indicate, as one would expect, that large variations occur, depending upon the characteristics of the air cleaner. Based on this available information, it is extremely difficult to estimate a typical ingression particle size distribution.

The final and most important information required to establish engine filtration requirements is the sensitivity of the critical engine components to contaminant wear. The data necessary must include an estimation of the sensitivity (wear rate) of engines with respect to both various particle size ranges and concentrations. McClelland and Billett presented a table of the relative effect of various particle size ranges upon engine wear. These tests were conducted with the addition of classified AC Fine Test Dust without a lube oil filter. Data

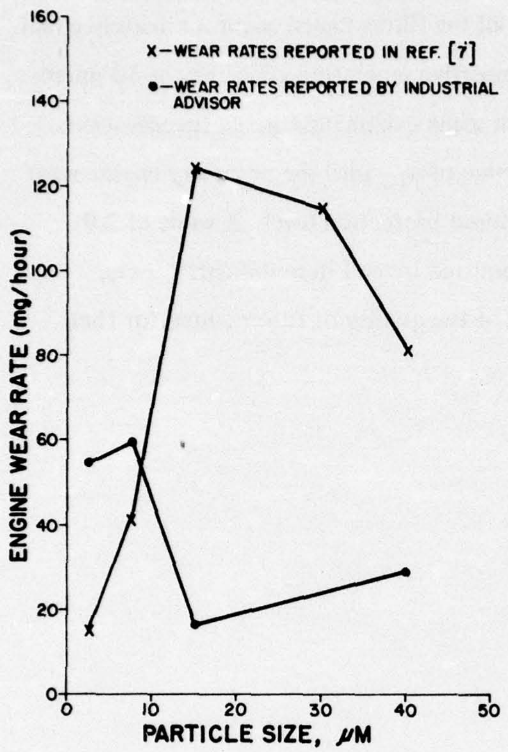

Fig. 4-1. Engine Wear Rates Vs. Particle Size.

were also revealed to OSU by an industrial advisor which present the same basic results; however, in these tests, a *"so called"* 40 micron absolute filter was utilized. Fig. 4-1 shows graphically the results of these two test programs where the particle size scale represents the mid-range of the particle size interval injected. In one case, the maximum rate occurred for the 10-20 micrometre injection; and, in the other instance, the maximum was at 5-10 micrometres.

The contamination level utilized, however, was not made available for either of the studies. In order to accurately define the engine contaminant sensitivity in terms compatible with filtration requirements, the effect of various particle concentrations of different particle sizes must be known. This type of information was not located in the literature or in the data supplied by industrial advisors.

Because of the lack of specific data relative to engine contaminant sensitivity, it is impossible to determine an accurate estimation of the filtration levels required for typical MERDC on-off highway diesel engine driven equipment. The proposed specification level given in Ref. [1] was a minimum average filtration ratio of 3.2 at 10 micrometres. Based upon the 48 multi-pass tests conducted on 35 different models of lube oil filters and reported in Tables 3-1 and 3-2, it appears that a minimum average α_{10} ratio of 3.2 is higher than

is currently being utilized for on-off highway equipment, as 71% of the filters tested had average α_{10} values less than 3.2. The median for all the filters tested is approximately equal to an α_{10} value of 2.0. This corresponds to a cumulative separation efficiency at 10 micrometres of 50%. This value seems to be in line with some existing industrial specifications, and it is recommended that MERDC utilize this value of α_{10} ultil the necessary engine wear data are available to accurately determine the required protection level. A value of 2.0 also appears to be the median for those filters submitted by end-item industrial users, which is probably a more accurate representation of the quality of filters suited for their equipment.

CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

The primary objective of the MERDC-OSU Lube Oil Filter Program was to develop an entire set of test procedures and specifications for lube oil filters which were industrially acceptable and compatible with U.S. Army MERDC requirements. Specifically, the direction of this year's effort was towards the establishment of the repeatability and reproducibility of the multi-pass test method in order to gain industrial acceptance. In addition, all available information was to be scanned in an attempt to establish filtration requirements based on engine wear data. It is believed that the objectives of the lube oil filter program were met as delineated in the preceding chapters.

A total of 48 filter tests utilizing the multi-pass method have been conducted at OSU and two other laboratories to establish the repeatability, reproducibility, and discriminatory characteristics of the test procedure. The results of most of these data have been revealed to industry either through handouts and discussions at the SAE lube oil filter test methods subcommittee or through the presentation of a technical paper at an SAE national convention. These presentations were made to familiarize industry with the test method and to help gain industrial acceptance. In addition, a presentation is scheduled before the Regular Common Carriers Maintenance Conference later in 1976. The SAE subcommittee is currently considering the adoption of the procedure as an industrial standard.

A proposed specification is included in Appendix C for MERDC equipment lube oil filters. The primary specification value for the particle separation characteristics (filtration ratio) resulting from the multi-pass test is proposed as 2.0 minimum for the $10\mu M$ size. An attempt was made to accurately specify the filtration level required based upon existing engine

wear data; however, sufficient data to make such a determination was not available. The minimum average α_{10} value of 2.0 was thus determined from the median of the tests conducted on the filters which were representative of those currently being utilized for on-off highway diesel engine driven vehicles.

It is recommended that, in order to receive maximum benefit from the MERDC-OSU lube oil studies, additional test work be performed on diesel engine contaminant sensitivity. These tests must be precisely controlled to determine the relative sensitivity to both various particle sizes and concentrations. One of the initial requirements of such a study would be to determine which parameter to measure for an indication of engine wear or performance degradation. It would be preferable that the engine not be disassembled between contaminant injections to eliminate possible influences.

A primary advantage which could be gained from such a contaminant sensitivity evaluation is the determination of whether filters utilized on hydraulic systems could be used interchangeably with lube oil filters. The economic factors of such interchangeability are obvious in the quantity purchasing, storage, and maintenance advantages. The authors of this report can see no obvious reason why such a specification could not be made; however, more specific data are required for accurate determinations.

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APPENDIX A

LUBE OIL FILTER TEST DATA

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 328C TEST LOCATION FPRC-OSU DATE 4-14-75

TEST FLOW [illegible] BUBBLE POINT NA INITIAL CLEANLINESS 12.6

TERMINAL ΔP [illegible] HOUSING ΔP 1.2 CLEAN ASSEMBLY ΔP 5.65 CLEAN ELEMENT ΔP 4.45

NET Δ P ([illegible])	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	[illegible]	7.4	9.2	12.8	19.9	34.7	41.2
Time (min.)	33.9	47.5	57	63	70	76.8	79.1

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.250	.252	251
Gravimetric Level (mg/litre)	3226	3112	[illegible]

BASE UPSTREAM LEVEL: 30 mg/litre

FINAL GRAVIMETRIC LEVEL: 25 mg/litre

ACCTD CAPACITY: 57.2 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5%	5066.1 / [illegible]	[illegible]	723.7 / 17.48	40.2	265.3 / 1.08	246	103.2 / .2	515	48.32 / .06	805
5%	[illegible] / [illegible]	1.52	656.3 / 10.4	63.1	216.9 / .88	246	86.32 / .22	392	38.28 / 0.14	273
10%	[illegible] / 376.1	[illegible]	654 / 14.72	43.8	217.6 / 1.22	178	88.88 / .36	247	43.48 / 0.06	725
20%	[illegible] / [illegible]	1.77	598.4 / 15.64	38.3	192.6 / 1.2	161	75.76 / .26	291	32.46 / 0.12	271
80%	[illegible] / [illegible]	[illegible]	567.02 / 16.16	35.1	185 / 2.18	84.9	72.38 / 1.0	72.4	33.04 / 0.34	97.2
ARITHMETIC AVERAGED		[illegible]		44.1		183		303		434

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 328G TEST LOCATION [illegible] DATE 4-24-75

TEST FLOW 7GPM BUBBLE POINT NA INITIAL CLEANLINESS [illegible]

TERMINAL ΔP 40 HOUSING ΔP 1.2 CLEAN ASSEMBLY ΔP [illegible] CLEAN ELEMENT ΔP [illegible]

NET Δ P (35.2)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	6.9	7.8	9.5	13.0	[illegible]	34.2	[illegible]
Time (min.)	43.0	53.7	60.2	65.4	71.1	76.2	78.2

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.246	.252	.249
Gravimetric Level (mg/litre)	2640	2496	2568

BASE UPSTREAM LEVEL: 24.1 mg/litre

FINAL GRAVIMETRIC LEVEL: 26 mg/litre

ACCTD CAPACITY: 50 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5%	3481.6 / 825.1	4.22	550.9 / 36.9	14.9	185.2 / 4.96	37.3	78.4 / 1.72	45.6	39.1 / .34	115
5%	3148.5 / 492.9	6.39	533.2 / 18.3	29.1	173.6 / 2.36	73.6	69.9 / .64	109	31.9 / .22	145
10%	3209.7 / 511.6	6.27	551.9 / 17.8	31.0	185.0 / 1.28	145	77.9 / .32	243	37.7 / .08	472
20%	3199.3 / 511.4	6.26	552.9 / 22.8	24.2	183.8 / 1.74	106	74.8 / .32	234	35.4 / .14	253
80%	3490.8 / 636.6	5.48	603.2 / 26.5	22.8	196.1 / 1.64	120	80.0 / .34	235	38.4 / .08	480
ARITHMETIC AVERAGED α		5.72		24.4		96.4		173		293

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 328 I TEST LOCATION EPRC-OSU DATE 1-24-75

TEST FLOW 7 GPM BUBBLE POINT NA INITIAL CLEANLINESS 8.6

TERMINAL ΔP 40 HOUSING ΔP 6.0 CLEAN ASSEMBLY ΔP 1.2 CLEAN ELEMENT ΔP 48

NET Δ P 35.2	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	6.9	7.8	9.5	13.0	20.1	34.2	41.2
Time (min.)	41.9	52.4	59.8	65.6	71.7	76.6	78.5

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.248	.252	.250
Gravimetric Level (mg/litre)	2530	2184	2357

BASE UPSTREAM LEVEL: 22.2 mg/litre

FINAL GRAVIMETRIC LEVEL: 27.5 mg/litre

ACCTD CAPACITY: 46.3 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10µM	α_{10}	> 20µM	α_{20}	> 30µM	α_{30}	> 40µM	α_{40}	> 50µM	α_{50}
2.5%	3310.7 / 648.3	5.11	522.7 / 16.7	31.3	168 / 1.7	99.0	67.7 / 0.42	161	30.4 / 0.4	76.0
5%	3152.2 / 461.2	6.83	538.1 / 12.6	42.7	178.1 / 0.86	207	73.8 / 0.2	369	33.0 / .08	413
10%	3172.4 / 562.7	5.64	547.5 / 23.0	23.8	181.6 / 1.88	96.6	74.0 / 0.5	148	33.2 / .26	128
20%	3254.6 / 558.8	5.82	555.6 / 23.3	23.8	181.2 / 1.82	100	72.3 / .36	201	33.5 / .22	152
80%	3348.3 / 715.7	4.68	558.9 / 35.6	15.7	181.1 / 3.1	58.4	73.5 / .78	94.2	34.2 / .3	114
ARITHMETIC AVERAGED α		5.62		27.5		112		195		177

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 328 J TEST LOCATION EPRC-OSU DATE 4-22-75

TEST FLOW 7 GPM BUBBLE POINT NA INITIAL CLEANLINESS 6.4 P/ml >10µM

TERMINAL ΔP 40 HOUSING ΔP 1.2 CLEAN ASSEMBLY ΔP 5.65 CLEAN ELEMENT ΔP 4.45

NET Δ P 35.55	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	6.5	7.4	9.2	12.8	19.9	34.7	41.2
Time (min.)	42	51.5	60.5	66.5	74	81.2	83.6

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.250	.250	.250
Gravimetric Level (mg/litre)	2992	2336	2664

BASE UPSTREAM LEVEL: 25.1 mg/litre

FINAL GRAVIMETRIC LEVEL: 26 mg/litre

ACCTD CAPACITY: 55.9 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10µM	α_{10}	> 20µM	α_{20}	> 30µM	α_{30}	> 40µM	α_{40}	> 50µM	α_{50}
2.5%	3716.3 / 596.8	6.23	675.6 / 20.36	33.2	236.6 / 1.28	185	102.4 / 0.26	394	52.2 / 0.1	522
5%	3467 / 488.4	7.10	652.1 / 24.3	26.8	223.6 / 2.64	84.7	95.4 / 0.76	126	47.4 / 0.3	159
10%	3482.2 / 451.8	7.71	659.8 / 18.48	35.7	225.9 / 1.38	164	97.3 / 0.24	405	48.5 / 0.12	405
20%	3332.1 / 408.7	8.15	628.1 / 19.66	31.9	212.4 / 1.40	152	91.5 / 0.16	572	47.7 / .08	597
80%	3555.7 / 406.1	8.76	692.2 / 25.16	27.5	236.1 / 2.0	118	101.8 / 0.32	318	50.5 / 0.2	253
ARITHMETIC AVERAGED α		7.59		31.0		141		363		387

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 330 B TEST LOCATION FPRC - OSU DATE 10-21-74

TEST FLOW 15 GPM BUBBLE POINT N/A INITIAL CLEANLINESS 122.3 #/ml > 10 μM

TERMINAL ΔP 40 psid HOUSING ΔP 6.9 CLEAN ASSEMBLY ΔP 7.0 CLEAN ELEMENT ΔP 0.1

NET Δ P (39.9)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P		9.0	11.0	15.0	23.0	38.9	46.9
Time (min.)		140.0	145.5	148.9	151.6	155.0	156.5

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.240	.265	.252
Gravimetric Level (mg/litre)	4668	3594	4131

BASE UPSTREAM LEVEL: 18.33 mg/litre

FINAL GRAVIMETRIC LEVEL: 88 mg/litre

ACCTD CAPACITY: 162.9 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 5 μM	α_5	> 10 μM	α_{10}	> 20 μM	α_{20}	> 25 μM	α_{25}	> 30 μM	α_{30}
2.5%	—		—		—		—		—	
5%	—		—		—		—		—	
10%	58668 / 53247	1.1	10097 / 8137	1.24	660 / 301	2.19	277 / 77.3	3.68	135 / 16.0	8.46
20%	58515 / 56795	1.03	10607 / 9811	1.08	655 / 413	1.59	232 / 94.7	2.45	103 / 18.7	5.58
40%	63794 / 51857	1.23	12207 / 9329	1.31	752 / 399	1.89	281 / 90.7	3.10	119 / 22.7	5.26
80%	41931 / 34321	1.22	8499 / 6387	1.33	606 / 309	1.96	226 / 79.3	2.85	109 / 18.0	6.04
ARITHMETIC AVERAGE α		1.15		1.24		1.91		3.00		6.37
TIME WEIGHTED AVERAGED α		1.10		1.24		2.18		3.53		8.29

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 329 B TEST LOCATION FPRC - OSU DATE 1-17-75

TEST FLOW 7 GPM BUBBLE POINT Spin-ON INITIAL CLEANLINESS 12.0

TERMINAL ΔP 40 HOUSING ΔP 1.2 CLEAN ASSEMBLY ΔP 7.4 CLEAN ELEMENT ΔP 6.2

NET Δ P (33.8)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	8.3	9.1	10.8	14.2	20.9	34.4	41.2
Time (min.)	42.8	68.9	89.0	100.4	109.8	119.1	122.6

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.249	.254	.252
Gravimetric Level (mg/litre)	2340	1868	2104

BASE UPSTREAM LEVEL: 20 mg/litre

FINAL GRAVIMETRIC LEVEL: 12 mg/litre

ACCTD CAPACITY: 65 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5%	2782 / 429	6.48	489 / 13.7	35.7	163 / 1.05	155	64.1 / 0.28	229	30.3 / 0.10	303
5%	3042 / 452	6.73	523 / 17.5	29.9	169 / 1.58	107	64.3 / 0.54	119	29.6 / 0.29	102
10%	2915 / 384	7.59	498 / 13.9	35.8	158 / 1.08	146	59.2 / 0.22	269	24.0 / 0.10	240
20%	2673 / 259	10.3	486 / 13.3	36.5	150 / 1.48	101	54.2 / 0.44	123	22.9 / 0.20	113
80%	2909 / 275	10.6	536 / 13.1	40.9	165 / 1.34	123	61.1 / 0.28	218	26.3 / 0.18	146
ARITHMETIC AVERAGED α		8.34		35.8		126		192		181

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 331 A TEST LOCATION FPRC-OSU DATE 2-26-76

TEST FLOW 10 GPM BUBBLE POINT 3.5" H_2O INITIAL CLEANLINESS 12.7/ml >10µm

TERMINAL ΔP 40 HOUSING ΔP 1.65 CLEAN ASSEMBLY ΔP 1.9 CLEAN ELEMENT ΔP 0.25

NET Δ P (39.25)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	2.9	3.9	5.9	9.9	17.8	33.7	41.7
Time (min.)	46.8	51.6	53.5	55.7	58.0	62.1	64.0

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	0.500	0.500	0.500
Gravimetric Level (mg/litre)	2019	1855	1937

BASE UPSTREAM LEVEL: 25.6 mg/litre

FINAL GRAVIMETRIC LEVEL: 37.0 mg/litre

ACCTD CAPACITY: 62.0 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10µM	α_{10}	> 20µM	α_{20}	> 30µM	α_{30}	> 40µM	α_{40}	> 50µM	α_{50}
2.5% UP / DOWN	25121 / 20386	1.23	1513 / 757	2.0	251 / 61.2	4.10	70.4 / 2.8	25.14	29.2 / 0.4	73.0
5.0% UP / DOWN	26169 / 23976	1.09	1832 / 1475	1.24	270 / 240	1.13	44.8 / 16.0	2.8	12.4 / 4.8	2.58
10.0% UP / DOWN	26975 / 24020	1.12	2300 / 1574	1.46	635 / 116	5.47	84.4 / 8.0	10.55	23.6 / 0.8	29.5
20.0% UP / DOWN	22562 / 19442	1.16	2218 / 1687	1.32	854 / 404	2.11	76.8 / 12.8	6.0	28.4 / 1.6	17.75
80.0% UP / DOWN	7652 / 4256	1.8	1014 / 363	2.79	420 / 77.2	5.44	81.6 / 4.0	20.4	33.2 / 0.4	83.0
ARITHMETIC AVERAGED α		1.28		1.76		3.65		12.78		37.62

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 332 A TEST LOCATION FPRC-OSU DATE 3-14-75

TEST FLOW 10 gpm BUBBLE POINT 8" H_2O INITIAL CLEANLINESS 6.6/ml >10µm

TERMINAL ΔP 40 HOUSING ΔP 2.8 CLEAN ASSEMBLY ΔP 3.1 CLEAN ELEMENT ΔP 0.3

NET Δ P (39.7)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	4.1	5.1	7.1	11.0	19.0	34.9	42.8
Time (min.)	75.0	81.5	86.0	89.7	94.2	99.7	102.1

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.250	.250	.250
Gravimetric Level (mg/litre)	4192	3252	3722

BASE UPSTREAM LEVEL: 25.6 mg/litre

FINAL GRAVIMETRIC LEVEL: 44.0 mg/litre

ACCTD CAPACITY: 95.0 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10µM	α_{10}	> 20µM	α_{20}	> 30µM	α_{30}	> 40µM	α_{40}	> 50µM	α_{50}
2.5% UP / DOWN	34776 / 27326	1.27	1562.2 / 345.4	4.52	354 / 12.58	28.4	120 / 0.88	136	48.5 / 0.40	122
5.0% UP / DOWN	17104.1 / 12133.4	1.41	922 / 251.4	3.67	210.7 / 9.92	20.3	71.1 / 0.40	178	30.1 / 0.04	753
10.0% UP / DOWN	18819.4 / 14106	1.33	1185 / 447.1	2.65	234.5 / 21.2	11.1	82.0 / 1.08	75.9	36.7 / 0.16	229
20.0% UP / DOWN	181772 / 14750	1.23	1246 / 592.5	2.10	219.8 / 29.6	7.43	71.5 / 1.36	52.6	29.0 / 0.24	121
80.0% UP / DOWN	10817.7 / 6117.7	1.76	1065 / 327.6	1.63	228.8 / 29.6	7.72	78.2 / 1.52	51.4	32.5 / 0.24	135
ARITHMETIC AVERAGED α		1.40		2.91		15.0		98.8		272

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 333A TEST LOCATION FPRC-OSU DATE 8-27-74

TEST FLOW 20GPM BUBBLE POINT N/A INITIAL CLEANLINESS 188.2 P/ml >10µm

TERMINAL ΔP 40PSID HOUSING ΔP 9.9 CLEAN ASSEMBLY ΔP 10.6 CLEAN ELEMENT ΔP 0.7

NET Δ P (39.3)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	11.6	12.6	14.6	18.5	26.3	42.0	49.9
Time (min.)	10.1	17.0	24.4	29.5	31.7	33.2	33.8

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.246	.258	.252
Gravimetric Level (mg/litre)	7840	7338	7589

BASE UPSTREAM LEVEL: 25.3 mg/litre

FINAL GRAVIMETRIC LEVEL: 75 mg/litre

ACCTD CAPACITY: 64.6 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10µM	α_{10}	> 15 µM	α^{15}	> 20 µM	α^{20}	> 30 µM	α^{30}	> 40 µM	α^{40}
2.5% UP / DOWN	8912 / 6369	1.37	17718 / 593	3.29	632 / 86	7.33	171.2 / 4.52	79.6	71.2 / 0.96	74.2
5.0% UP / DOWN	9997 / 4784	2.09	1854 / 466	3.98	644 / 70	9.22	161.9 / 2.24	72.1	56.8 / 0.16	355
10.0% UP / DOWN	10389 / 6326	1.64	1965 / 621	3.17	669 / 116	5.77	175 / 3.70	47.6	68.8 / 0.64	108
20.0% UP / DOWN	11051 / 7130	1.55	2140 / 805	2.66	707 / 125	5.69	184.5 / 5.12	36.0	65.6 / 1.28	52.3
80.0% UP / DOWN	16143 / 12648	1.28	3236 / 1922	1.68	942 / 342	2.75	186.3 / 16.96	11.6	69.3 / 1.44	48.1
ARITHMETIC AVERAGED α		1.58		2.96		6.10		41.4		118.5

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 333B TEST LOCATION FPRC-OSU DATE 8-28-74

TEST FLOW 20GPM BUBBLE POINT N/A INITIAL CLEANLINESS 9.5 P/ml >10µm

TERMINAL ΔP 40PSID HOUSING ΔP 9.9 CLEAN ASSEMBLY ΔP 10.6 CLEAN ELEMENT ΔP 0.7

NET Δ P (39.3)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	11.6	12.6	14.6	18.5	26.3	42.0	49.9
Time (min.)	5.0	9.1	13.3	16.5	17.9	18.5	18.7

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.500	.498	.499
Gravimetric Level (mg/litre)	6600	6300	6450

BASE UPSTREAM LEVEL: 47.5 mg/litre

FINAL GRAVIMETRIC LEVEL: 187 mg/litre

ACCTD CAPACITY: 60.2 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10µM	α_{10}	> 15 µM	α^{15}	> 20 µM	α^{20}	> 30 µM	α^{30}	> 40 µM	α^{40}
2.5% UP / DOWN	12542 / 6761	1.85	2731 / 807	3.39	1026 / 132	7.75	273 / 8.8	31.0	100.6 / 1.60	63.0
5.0% UP / DOWN	14762 / 8408	1.76	3006 / 884	3.40	1072 / 130	8.27	282 / 6.0	46.9	102.8 / 1.2	85.9
10.0% UP / DOWN	—		—		—		—		—	
20.0% UP / DOWN	18914 / 12148	1.56	3672 / 1430	2.57	1245 / 238	5.23	305 / 7.2	42.3	112 / 0.8	140
80.0% UP / DOWN	26921 / 20601	1.31	5179 / 3057	1.69	1471 / 561	2.62	290 / 228	12.7	111.6 / 1.20	93
ARITHMETIC AVERAGED α		1.62		2.76		5.97		33.2		95.9

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 333C TEST LOCATION FPBC-OSU DATE 8-26-74

TEST FLOW 20GPM BUBBLE POINT N/A INITIAL CLEANLINESS 5.6 P/ml >10µm

TERMINAL ΔP 40PSID HOUSING ΔP 9.9 CLEAN ASSEMBLY ΔP 10.6 CLEAN ELEMENT ΔP 0.7

NET Δ P (39.3)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	1.0	2.0	3.9	7.9	15.7	31.4	39.3
Time (min.)	21.3	38.5	61.8	79.5	87.7	93.5	94.9

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.250	.240	.245
Gravimetric Level (mg/litre)	2786	2438	2612

BASE UPSTREAM LEVEL: 8.45 mg/litre

FINAL GRAVIMETRIC LEVEL: 5.0 mg/litre

ACCTD CAPACITY: 60.7 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10µm	α 10	> 15µm	α 15	> 20µm	α 20	> 30µm	α 30	> 40µm	α 40
2.5% UP / DOWN	18314 / 14451	1.27	2833 / 1389	2.04	259 / 31.5	8.21	72.0 / 1.30	56.4	26.6 / 0.4	69.0
5.0% UP / DOWN	13764 / 11180	1.23	2236 / 1161	1.93	214 / 26.9	7.93	61.2 / 2.70	47.1	25.0 / .10	22.7
10.0% UP / DOWN	11088 / 7093	1.84	2054 / 848	2.42	221 / 25.1	8.74	61.7 / 2.70	22.9	22.0 / 0.10	220
20.0% UP / DOWN	9621 / 4891	1.98	1932 / 653	2.96	246 / 21.2	11.6	71.3 / 1.30	54.8	26.5 / 0.30	88.3
80.0% UP / DOWN	9173 / 5773	1.68	2074 / 802	2.58	234 / 40.7	5.75	60.0 / 2.10	28.6	23.9 / 0.30	79.7
ARITHMETIC AVERAGED α		1.62		2.38		8.45		41.8		94.9

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 334C TEST LOCATION FPRC-OSU DATE 4-25-75

TEST FLOW 20GPM BUBBLE POINT NA INITIAL CLEANLINESS 3.9

TERMINAL ΔP 40 HOUSING ΔP 6.0 CLEAN ASSEMBLY ΔP 6.1 CLEAN ELEMENT ΔP 0.1

NET Δ P (39.9)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	7.1	8.1	10.1	14.1	22.1	38.0	46.0
Time (min.)	28.3	32.0	34.3	35.8	37.2	39.2	40.2

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.246	.252	.249
Gravimetric Level (mg/litre)	9544	7804	8674

BASE UPSTREAM LEVEL: 28.5 mg/litre

FINAL GRAVIMETRIC LEVEL: 36 mg/litre

ACCTD CAPACITY: 86.8 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10µm	α_{10}	> 20µm	α_{20}	> 30µm	α_{30}	> 40µm	α_{40}	> 50µm	α_{50}
2.5%	6522.1 / 3162.8	2.06	701.5 / 83.66	8.39	204.2 / 4.92	41.5	80.44 / .68	118	36.86 / .3	123
5%	5999.5 / 2526.8	2.37	703.5 / 80.62	8.73	205.0 / 3.56	57.6	82.32 / .50	165	38.38 / .16	240
10%	7479.7 / 4196.9	1.78	826.3 / 166.9	4.95	227.0 / 10.46	21.7	87.1 / .76	115	41.14 / .12	342
20%	8445.4 / 6080.4	1.66	927.2 / 246.9	3.76	231.5 / 16.62	13.9	87.44 / 1.32	66.2	40.14 / .10	401
80%	6874.5 / 3233.2	2.13	894.4 / 213.9	4.18	231.2 / 16.78	13.8	87.44 / 0.9	97.1	40.14 / .12	335
ARITHMETIC AVERAGED α		2.0		6.0		29.7		112.7		288

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 335A TEST LOCATION FPRC-OSU DATE 5-20-75

TEST FLOW 10GPM BUBBLE POINT 16" H_2O INITIAL CLEANLINESS 3.5/ml>10µm

TERMINAL ΔP 40PSID HOUSING ΔP 3.0 CLEAN ASSEMBLY ΔP 5.5 CLEAN ELEMENT ΔP 2.5

NET Δ P (37.5)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	6.44	7.38	9.25	13.0	20.5	35.5	43.0
Time (min.)	3.7	6.4	9.3	12.0	14.0	15.6	16.0

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.250	.254	.252
Gravimetric Level (mg/litre)	4692	4434	4563

BASE UPSTREAM LEVEL: 30.4 mg/litre

FINAL GRAVIMETRIC LEVEL: 44 mg/litre

ACCTD CAPACITY: 18.4 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10µM	α_{10}	> 20µM	α_{20}	> 30µM	α_{30}	> 40µM	α_{40}	> 50µM	α_{50}
2.5%	3071 / 97.8	31.4	642.0 / 20.7	31.0	220.9 / 7.13	31.0	90.7 / 2.85	31.8	41.02 / 1.30	31.6
5%	3175 / 105.1	30.2	648.2 / 21.8	29.7	214.7 / 7.46	28.7	86.4 / 3.38	25.6	38.6 / 1.82	21.2
10%	3256 / 119.2	27.3	419.8 / 27.1	15.5	222 / 9.90	22.4	88.1 / 4.62	19.1	39.8 / 2.34	17.0
20%	3402 / 183.	18.6	687.5 / 36.7	18.7	234.3 / 13.1	17.9	92.6 / 5.46	17.0	42.3 / 2.80	15.1
80%	3724 / 442.1	8.42	761.5 / 89.4	8.52	255 / 31.9	7.99	99.9 / 16.8	5.95	42.9 / 6.28	6.8
ARITHMETIC AVERAGED α		23.2		20.7		21.6		19.9		18.3

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 336A TEST LOCATION FPRC-OSU DATE 9-15-74

TEST FLOW 10GPM BUBBLE POINT 13 in. H_2O INITIAL CLEANLINESS 2155 - Rel. No/ml >10µ

TERMINAL ΔP 40psid HOUSING ΔP 4.5 CLEAN ASSEMBLY ΔP 6.2 CLEAN ELEMENT ΔP 1.7

NET Δ P (38.3)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P		8.1	10.0	13.9	21.5	36.8	44.5
Time (min.)		13.1	15.9	20.1	24.3	27.8	29.0

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.248	.260	.256
Gravimetric Level (mg/litre)	4328	3589	3959

BASE UPSTREAM LEVEL: 26.72 mg/litre

FINAL GRAVIMETRIC LEVEL: 26.6 mg/litre

ACCTD CAPACITY: 29.4 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 5 µM	α_5	> 10 µM	α_{10}	> 20 µM	α_{20}	> 30 µM	α_{30}	> 40 µM	α_{40}
2.5%	——		——		——		——		——	
5%	8726 / 578	15.1	2708 / 124	21.8	528.9 / 26	20.3	180.6 / 8.9	20.3	75.0 / 3.3	22.7
10%	8629 / 534	16.2	2690 / 113	23.8	514 / 23	22.4	166.2 / 6.9	24.1	64.3 / 2.6	24.7
20%	8337 / 1042	12.9	2578 / 149	17.3	509.4 / 28.5	17.9	164.2 / 9.4	17.5	64.9 / 3.4	19.1
40%	8553 / 1042	8.21	2644 / 236	11.2	521.3 / 45.1	11.6	169 / 17.3	9.67	69.2 / 6.8	10.2
80%	9808 / 1708	5.74	2921 / 370	7.91	578 / 80.6	7.17	191.1 / 31.8	6.01	77.0 / 14.6	5.2
ARITHMETIC AVERAGE α		11.6		16.4		15.8		15.5		16.4
TIME WEIGHTED AVERAGED α		12.7		18.0		17.8		16.9		18.3

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 337B TEST LOCATION FPRC-OSU DATE 10-29-74

TEST FLOW 12GPM BUBBLE POINT 4in H₂O INITIAL CLEANLINESS 8.9 $/ml > 10\mu m$

TERMINAL ΔP 40psi HOUSING ΔP 22.8 CLEAN ASSEMBLY ΔP 28.7 CLEAN ELEMENT ΔP 5.9

NET Δ P (34.1)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	29.6	30.4	32.1	35.5	42.3	56	62.8
Time (min.)	28.7	36.1	46.2	54.4	61.9	67.8	69.6

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.248	.250	.249
Gravimetric Level (mg/litre)	5186	4492	4839

BASE UPSTREAM LEVEL: 26.5 mg/litre

FINAL GRAVIMETRIC LEVEL: 25 mg/litre

ACCTD CAPACITY: 84 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 5 µM	α_5	> 10 µM	α_{10}	> 20 µM	α_{20}	> 30 µM	α_{30}	> 40 µM	α_{40}
2.5%	15660 / 11196	1.40	2853 / 146	19.6	541 / 6.4	84.6	188 / 2.2	85.6	82.6 / 1.0	82.6
5%	14006 / 8666	1.62	2596 / 115	22.7	487 / 0.7	696	168 / 0.0	∞	71.8 / 0.0	∞
10%	11347 / 5329	2.13	2479 / 92.3	26.9	459 / 0.5	918	158 / 0.1	1582	67.8 / 0.0	∞
20%	10711 / 4218	2.54	2851 / 102	27.8	463 / 0.5	927	153 / 0.2	763	65.8 / 0.2	329
40%	16844 / 3706	4.55	2493 / 120	20.7	446 / 0.6	744	152 / 0.2	753	67.8 / 0.0	∞
80%	10283 / 3220	3.19	2460 / 157	15.7	450 / 3.7	122	155 / 1.0	155	66.2 / 0.5	132
ARITHMETIC AVERAGE α		2.57		22.2		581		∞		∞
TIME WEIGHTED AVERAGED α		2.27		22.3		457		∞		∞

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 339A TEST LOCATION FPRC-OSU DATE 3-11-75

TEST FLOW 20GPM BUBBLE POINT 4.8" H₂O INITIAL CLEANLINESS 8.05 $/ml > 10\mu m$

TERMINAL ΔP 40 HOUSING ΔP 8.3 CLEAN ASSEMBLY ΔP 8.8 CLEAN ELEMENT ΔP 0.5

NET Δ P (39.5)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	9.8	10.8	12.8	16.7	24.6	40.4	48.3
Time (min.)	55.5	66	83.5	103	123.5	149	159

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.250	.248	.249
Gravimetric Level (mg/litre)	5461.3	6139.3	5800

BASE UPSTREAM LEVEL: 19.1 mg/litre

FINAL GRAVIMETRIC LEVEL: 18.4 mg/litre

ACCTD CAPACITY: 230 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10µM	α_{10}	> 20µM	α_{20}	> 30µM	α_{30}	> 40µM	α_{40}	> 50µM	α_{50}
2.5%	7103.6 / 5925.1	1.20	504.9 / 117.5	4.30	124.4 / 9.1	13.7	39.6 / 1.3	30.5	15.6 / 1.0	15.6
5%	2866.8 / 956	3.00	395 / 29.9	13.2	115.4 / 4.2	27.5	41.3 / 1.6	25.8	18.5 / 0.9	20.6
10%	2342.4 / 214.5	10.9	402.6 / 8.42	47.8	124.2 / 1.08	115	44.6 / 0.33	135	20.1 / 0.25	84
20%	2331.2 / 218.2	10.7	407.2 / 14.1	28.9	126.2 / 1.4	90.1	46.2 / 0.8	57.8	18.2 / 0.3	60.7
80%	2432.7 / 132	18.4	472.3 / 12.1	39.0	156.1 / 3.1	50.4	60.9 / 2.1	29	31.9 / 0.7	45.6
ARITHMETIC AVERAGED α		8.85		26.6		59.3		55.6		45.3

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 339A TEST LOCATION FPRC-OSU DATE 4-8-75

TEST FLOW 12 GPM BUBBLE POINT 8" H_2O INITIAL CLEANLINESS 2.6 part/ml > 10 μm

TERMINAL ΔP 40.0 HOUSING ΔP 21.0 CLEAN ASSEMBLY ΔP 25.7 CLEAN ELEMENT ΔP 4.7

NET Δ P (35.3)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	26.6	27.5	28.2	32.8	39.8	53.9	61.0
Time (min.)	11.9	22.3	25.7	28.3	29.9	31.5	32.3

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.246	.252	.248
Gravimetric Level (mg/litre)	5220	4174	4697

BASE UPSTREAM LEVEL: 25.6 mg/litre

FINAL GRAVIMETRIC LEVEL: 52 mg/litre

ACCTD CAPACITY: 37.6 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5%	9103.2 / 5615.2	1.62	705.2 / 9.2	76.7	316.5 / .48	451	85.3 / .14	610	37.8 / 0	∞
5%	6774.7 / 2640.8	2.57	673.1 / 5.46	123	213.4 / .36	593	83.4 / .14	596	35.7 / .06	595
10%	5882.4 / 1980	2.97	653.5 / 5.42	121	209.8 / .38	552	82.6 / .12	688	38.0 / .04	950
20%	6390.2 / 2396	2.67	693.7 / 10.12	68.5	217.8 / .38	573	84.7 / .08	1059	37.4 / .06	623
80%	6580.7 / 2584.2	2.55	717.3 / 22.26	32.2	220.3 / .58	380	87.1 / .16	544	37.7 / .06	628
ARITHMETIC AVERAGED α		2.48		84.3		510		699		∞

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 340F TEST LOCATION FPRC-OSU DATE 3-13-75

TEST FLOW 15 GPM BUBBLE POINT 3" H_2O INITIAL CLEANLINESS 9.8 P/ml > 10 μm

TERMINAL ΔP 40 HOUSING ΔP 5.6 CLEAN ASSEMBLY ΔP 5.7 CLEAN ELEMENT ΔP 0.1

NET Δ P (39.9)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	6.7	7.7	9.7	13.7	21.7	37.6	45.6
Time (min.)	77	84.3	88.5	91.6	95	100.1	102.4

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.250	.250	.250
Gravimetric Level (mg/litre)	5510	4160	4835

BASE UPSTREAM LEVEL: 21.3 mg/litre

FINAL GRAVIMETRIC LEVEL: 48 mg/litre

ACCTD CAPACITY: 123.8 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5%	21581.8 / 18156.1	1.19	1001.8 / 383.8	2.61	220.8 / 21.7	10.2	82.5 / 2.88	28.6	37.7 / 0.8	47.1
5%	21847.6 / 18257.6	1.20	1152.5 / 547.4	2.11	209 / 34.2	6.11	66.6 / 4.08	16.3	27.8 / 1.28	21.7
10%	26077 / 22547.8	1.16	1659.1 / 1085.8	1.53	241.2 / 76.8	3.14	65.9 / 6.16	10.7	27.5 / 1.68	16.4
20%	27035.8 / 22565.8	1.20	1991 / 1347.1	1.48	283 / 105.8	2.67	74.1 / 11.8	6.28	28.6 / 2.96	9.65
80%	6483.4 / 4890	1.33	909.04 / 381.0	2.39	206.6 / 47.7	4.33	69.7 / 9.52	7.32	29.6 / 1.60	18.5
ARITHMETIC AVERAGED α		1.22		2.02		5.29		13.8		22.7

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 341 A TEST LOCATION FPRC-OSU DATE 11-16-74

TEST FLOW 15 GPM BUBBLE POINT N/A INITIAL CLEANLINESS 172 Part/ml >10μM

TERMINAL ΔP 40PSID HOUSING ΔP 7.8 CLEAN ASSEMBLY ΔP 8.1 CLEAN ELEMENT ΔP 0.3

NET Δ P 39.7	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	9.1	22.3	12.1	16.0	24.0	39.9	47.8
Time (min.)	10.1	33.2	41.2	44.6	46.8	48.4	49.0

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.248	.250	.249
Gravimetric Level (mg/litre)	5802	5178	5490

BASE UPSTREAM LEVEL: 24.1 mg/litre

FINAL GRAVIMETRIC LEVEL: 71.0 mg/litre

ACCTO CAPACITY: 89.6 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 5 μM	α_5	> 10 μM	α_{10}	> 20 μM	α_{20}	> 30 μM	α_{30}	> 40 μM	α_{40}
2.5%	52559 / 42055	1.25	4702 / 2131	2.21	520 / 25	20.8	174 / 0	∞	73 / 0	∞
5%	46302 / 44784	1.03	4262 / 2239	1.9	515 / 25	20.6	160 / 1	160	72 / 1	72
10%	42153 / 32407	1.3	4534 / 1984	2.29	497 / 30	16.6	149 / 3	49.6	62 / 8	7.75
20%	49345 / 31664	1.56	5874 / 2772	2.12	562 / 38	14.8	176 / 0	∞	82 / 0	∞
40%	52282 / 38916	1.34	6937 / 4076	1.70	515 / 77	5.69	154 / 2	77	53 / 2	26.5
80%	65461 / 48494	1.35	9216 / 5369	1.72	672 / 117	5.74	178 / 5	89	70 / 2	35
ARITHMETIC AVERAGE α		1.31		1.99		14.0		∞		∞
TIME WEIGHTED AVERAGED α		1.16		1.99		18.0		∞		∞

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 342B TEST LOCATION FPRC DATE 3-17-75

TEST FLOW 28 GPM BUBBLE POINT 30+" H_2O INITIAL CLEANLINESS

TERMINAL ΔP 40 psid HOUSING ΔP 20 CLEAN ASSEMBLY ΔP 20.4 CLEAN ELEMENT ΔP 0.4

NET Δ P 39.6	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	21.4	22.4	24.4	28.3	36.2	52.1	60
Time (min.)	64.7	90.5	113	133	153	179	192

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.250	.250	.250
Gravimetric Level (mg/litre)	12734.	8424.	10579.

BASE UPSTREAM LEVEL: 24.96 mg/litre

FINAL GRAVIMETRIC LEVEL: 27.0 mg/litre

ACCTD CAPACITY: 507.3 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5%	2890. / 175.4	16.5	566 / 10.3	55.0	183 / 1.64	112.	68.9 / 0.56	123	28.6 / 0.23	124
5%	2704. / 111.6	24.2	535 / 6.02	88.9	173 / 0.80	216	64.8 / 0.54	120	28.4 / 0.14	203
10%	2626 / 76.88	34.2	509 / 4.74	107	162 / 0.68	239	58.1 / 0.32	182	22.9 / 0.16	143
20%	2586 / 52.5	49.3	521 / 3.50	149	166 / 0.22	752	60.8 / 0.02	3040	24.7 / 0.02	1234
80%	2526 / 43.4	58.2	505 / 3.72	136	164 / 0.38	433	61.8 / 0.06	1030	22.8 / 0	∞
ARITHMETIC AVERAGED α		36.5		107		350		899		∞

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 343 D TEST LOCATION FPRC-OSU DATE 9-25-74

TEST FLOW 28 GPM BUBBLE POINT > 30 in. H_2O INITIAL CLEANLINESS 715.4 P/ml >10 μm

TERMINAL ΔP 40 PSID HOUSING ΔP ____ CLEAN ASSEMBLY ΔP ____ CLEAN ELEMENT ΔP ____

NET Δ P (39.3)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	29.4	30.4	32.3	36.3	44.1	59.8	67.7
Time (min.)	19.7	23.8	36.5	41.0	44.5	47.0	47.8

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.250	.256	.253
Gravimetric Level (mg/litre)	12056	10843	11449

BASE UPSTREAM LEVEL: 27.3 mg/litre

FINAL GRAVIMETRIC LEVEL: 77.0 mg/litre

ACCTD CAPACITY: 138.5 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10 μM	α_{10}	> 15 μM	α_{15}	> 20 μM	α_{20}	> 30 μM	α_{30}	> 40 μM	α_{40}
2.5%	5760 / 2274	2.53	1477 / 266	5.56	591 / 58.7	10.1	184 / 16.0	11.5	77.5 / 9.60	8.08
5%	4716 / 1551	3.04	1284 / 182	7.05	539 / 26.7	20.2	166 / 2.00	83.0	73.8 / 0.40	185
10%	4319 / 1351	3.20	1234 / 173	7.14	518 / 30.5	17.0	154 / 2.56	60.3	65.6 / 0.96	68.3
20%	4281 / 1363	3.14	1252 / 200	6.27	527 / 39.9	13.2	157 / 4.48	35.0	63.6 / 1.12	56.8
40%	4506 / 1647	2.74	1326 / 249	5.33	556 / 43.4	12.8	165 / 2.48	66.7	65.7 / 0.16	411
80%	5668 / 2639	2.14	1529 / 396	3.86	572 / 75.3	7.59	159 / 7.04	22.6	63.7 / 1.76	36.2
ARITHMETIC AVERAGE α		2.80		5.87		13.5		46.5		127
TIME WEIGHTED AVERAGED α		2.81		6.08		13.2		37.9		76.0

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 344B TEST LOCATION FPRC-OSU DATE 11-13-77

TEST FLOW 24 GPM BUBBLE POINT 9 in. H_2O INITIAL CLEANLINESS 22.1 P/ml ≥10 μm

TERMINAL ΔP 40 PSID HOUSING ΔP 19.7 CLEAN ASSEMBLY ΔP 20.2 CLEAN ELEMENT ΔP 0.5

NET Δ P ( )	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	21.2	22.2	24.1	28.1	36.0	51.8	59.7
Time (min.)	35.0	38.0	44.8	46.0	47.1	48.5	49.1

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.250	.242	.246
Gravimetric Level (mg/litre)	10700	9248	9974

BASE UPSTREAM LEVEL: 26.6 mg/litre

FINAL GRAVIMETRIC LEVEL: 130 mg/litre

ACCTD CAPACITY: 120.5 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 30 μM	α_{30}	> 40 μM	α_{40}	> 50 μM	α_{50}	> 60 μM	α_{60}	> 70 μM	α_{70}
2.5%	294 / 96.6	3.04	74.2 / 5.68	13.1	29.9 / 0.68	44.0	19.8 / 0.40	49.6	9.4 / 0.16	58.8
5%	349 / 178	1.96	79.7 / 11.9	6.68	32.8 / 9.68	3.38	21.6 / 1.08	20.0	11.0 / 0.48	22.9
10%	506 / 393	1.29	88.3 / 29.4	3.00	31.0 / 4.80	6.47	19.9 / 1.72	11.6	9.72 / 0.68	14.3
20%	774 / 617	1.25	122 / 56.0	2.18	38.8 / 8.20	4.74	22.6 / 2.84	7.96	10.9 / 1.00	10.9
40%	785 / 577	1.36	129 / 59	2.18	38.3 / 9.6	3.99	23.2 / 3.48	6.68	11.0 / 1.04	10.6
80%	514 / 274	1.88	101 / 31.6	3.20	35.1 / 5.44	6.45	20.8 / 2.52	8.24	9.52 / 0.92	10.3
ARITHMETIC AVERAGE α		1.80		5.06		11.5		19.3		21.3
TIME WEIGHTED AVERAGED α		2.60		10.5		33.2		39.2		46.6

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 345A TEST LOCATION FPRC-OSU DATE April 3-1975

TEST FLOW 24 GPM BUBBLE POINT 7" H_2O INITIAL CLEANLINESS 3.3 part/ml >10µm

TERMINAL ΔP 40 HOUSING ΔP 14.4 CLEAN ASSEMBLY ΔP 14.6 CLEAN ELEMENT ΔP .2

NET Δ P (35.8)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	15.595	16.59	18.58	22.56	30.52	46.4	54.4
Time (min.)	33.7	40.8	45.2	49.4	53.6	58.8	61.0

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.247	.255	.251
Gravimetric Level (mg/litre)	11756	8962	10354

BASE UPSTREAM LEVEL: 17.2 mg/litre

FINAL GRAVIMETRIC LEVEL: 56 mg/litre

ACCTD CAPACITY: 158.5 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10µM	α_{10}	> 20µM	α_{20}	> 30µM	α_{30}	> 40µM	α_{40}	> 50µM	α_{50}
2.5%	7358.6 / 3527.3	2.09	760.8 / 76.86	9.90	213.5 / 5.02	42.5	78.94 / 1.12	70.5	32.36 / .48	67.4
5%	5084.5 / 1732.9	2.93	683.8 / 57.88	11.81	204.2 / 5.02	40.7	78.1 / 1.4	55.8	35.24 / .52	67.8
10%	4294 / 1135.4	3.78	671.3 / 51.78	13.0	205 / 5.30	38.7	82.4 / 1.3	63.4	38.26 / .56	68.3
20%	4176.7 / 1013.2	4.12	685.6 / 60.04	11.4	213.8 / 6.8	31.4	81.68 / 1.58	51.4	36.3 / .72	50.4
80%	4022.9 / 942.6	4.27	561.7 / 70.66	7.95	206.6 / 10.42	19.8	77.4 / 3.24	23.9	33.3 / 1.3	25.6
ARITHMETIC AVERAGED α		3.44		10.8		34.6		53.0		55.9

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 346A TEST LOCATION FPRC-OSU DATE 3-12-74

TEST FLOW 11.5 GPM BUBBLE POINT NA INITIAL CLEANLINESS 13 particles/ml >10µm

TERMINAL ΔP 40 PSI HOUSING ΔP 11.0 CLEAN ASSEMBLY ΔP 11.4 CLEAN ELEMENT ΔP 0.4

NET Δ P (39.6)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	12.4	13.4	15.4	19.3	27.2	43.1	51
Time (min.)	26	33.4	37	38.8	40.3	41.6	42

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.250	.250	.250
Gravimetric Level (mg/litre)	5080	4352	4716

BASE UPSTREAM LEVEL: 27.1 mg/litre

FINAL GRAVIMETRIC LEVEL: 105 mg/litre

ACCTD CAPACITY: 49.5 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10µM	α_{10}	> 20µM	α_{20}	> 30µM	α_{30}	> 40µM	α_{40}	> 50µM	α_{50}
2.5%	10617.6 / 9251.6	1.15	661.6 / 253.6	2.61	158 / 20	7.9	56.4 / 2.4	23.5	28.4 / 0.4	71
5%	10090.8 / 8740.4	1.16	693.6 / 307.6	2.25	162.4 / 30	5.41	65.2 / 6.8	9.59	26.8 / 2.4	11.2
10%	10816.8 / 9666.8	1.12	759.6 / 408	1.86	171.6 / 40.8	4.21	55.6 / 4.4	12.6	20.8 / 0	∞
20%	12759.6 / 10662.4	1.20	1132 / 544.4	2.08	252 / 56	4.49	95.2 / 4.8	19.8	50.4 / 0.4	126
80%	10313 / 7049.6	1.46	953.2 / 459.6	2.07	206.8 / 68.4	3.02	67.6 / 8.0	8.45	35.2 / 2.4	14.7
ARITHMETIC AVERAGED α		1.22		2.17		5.01		14.8		∞

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 347A TEST LOCATION FPRC-OSU DATE 3-12-74

TEST FLOW 11.5GPM BUBBLE POINT NA INITIAL CLEANLINESS 14.1 part/ml >10µm

TERMINAL ΔP 40PSID HOUSING ΔP 15.2 CLEAN ASSEMBLY ΔP 15.6 CLEAN ELEMENT ΔP 0.4

NET Δ P (39.6)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	16.6	17.6	19.6	23.5	31.4	47.8	55.2
Time (min.)	19	31	35.7	37.7	39	*	

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.248	.252	.250
Gravimetric Level (mg/litre)	4048	3930	3989

BASE UPSTREAM LEVEL: 22.9 mg/litre

FINAL GRAVIMETRIC LEVEL: 213 mg/litre

ACCTD CAPACITY: 38.9 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10µM	α_{10}	> 20µM	α_{20}	> 30µM	α_{30}	> 40µM	α_{40}	> 50µM	α_{50}
2.5%	28391.5 / 26209.8	1.08	1801.4 / 1135.8	1.59	368 / 172.5	2.13	122 / 64.3	1.90	53.0 / 33.1	1.60
5%	30299.2 / 26105.4	1.16	1669 / 998.9	1.67	299.2 / 78.8	3.80	100.7 / 8.0	12.6	49.2 / 2.16	19.1
10%	30867.8 / 28709.0	1.08	2038.5 / 1434.6	1.42	331.9 / 133.8	2.48	94.2 / 19.8	4.76	41.3 / 3.76	11.0
20%	34902 / 30262	1.15	2733.7 / 2042.3	1.34	423.0 / 216.8	1.95	109 / 27.2	4.01	41.5 / 4.16	9.98
* 80%	—		—		—		—		—	
ARITHMETIC AVERAGED α		1.12		1.51		2.59		5.82		10.4

* RELIEF VALVE OPENED

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 350B TEST LOCATION FPRC-OSU DATE 11-22-74

TEST FLOW 25GPM BUBBLE POINT N/A INITIAL CLEANLINESS 22.4 #/ml >10µm

TERMINAL ΔP 40PSID HOUSING ΔP 17.0 CLEAN ASSEMBLY ΔP 26.5 CLEAN ELEMENT ΔP 9.5

NET Δ P (30.5)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	27.3	28.0	29.6	32.6	38.7	50.9	57.0
Time (min.)	11.5	14.0	29.1	43.5	60.1	74.5	77.5

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.250	.252	.251
Gravimetric Level (mg/litre)	12321	8886	10603

BASE UPSTREAM LEVEL: 28.1 mg/litre

FINAL GRAVIMETRIC LEVEL: 61.8 mg/litre

ACCTD CAPACITY: 206 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 5 µM	α_5	> 10 µM	α_{10}	> 20 µM	α_{20}	> 30 µM	α_{30}	> 40 µM	α_{40}
2.5%	22203 / 11901	1.87	3634 / 835	4.35	524 / 48	10.9	156 / 5.25	29.8	63.8 / 2.0	31.9
5%	24298 / 11817	2.06	3718 / 703	5.29	543 / 25.	21.7	173 / 4.60	37.6	74.2 / 2.0	37.1
10%	23209 / 11374	2.04	3613 / 643	5.62	544 / 26.6	20.4	171 / 6.0	28.5	75.2 / 1.4	53.7
20%	23742 / 10931	2.17	3677 / 631	5.83	550 / 28	19.6	167 / 3.4	49.2	67.2 / 1.2	56.0
40%	21982 / 11472	1.92	3610 / 700	5.16	534 / 29.2	18.3	167 / 3.6	46.3	64.9 / 0.6	107
80%	18815 / 10192	1.85	3271 / 703	4.65	476 / 26.2	18.2	137 / 3.2	42.8	57.2 / 0.8	71.5
ARITHMETIC AVERAGE α		1.99		5.15		18.2		39.0		59.6
TIME WEIGHTED AVERAGED α		1.98		5.16		17.9		39.7		65.6

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 351 A TEST LOCATION FPRC-OSU DATE 4-14-75

TEST FLOW 25 GPM BUBBLE POINT NA INITIAL CLEANLINESS 4.2

TERMINAL ΔP 40 HOUSING ΔP 10.2 CLEAN ASSEMBLY ΔP 39.8 CLEAN ELEMENT ΔP 29.6

NET Δ P (10.4)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	40.1	40.3	40.8	41.9	43.96	48.1	50.2
Time (min.)	8.0	10.2	24.7	58.3	71.1	101.6	141

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.250	.254	.252
Gravimetric Level (mg/litre)	9812	9112	9462

BASE UPSTREAM LEVEL: 15.7 mg/litre

FINAL GRAVIMETRIC LEVEL: 494 mg/litre

ACCTD CAPACITY: 336 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5%	6690.1 / 2492	2.68	1366.2 / 630.6	2.17	468.5 / 134.8	3.47	195.4 / 99.0	1.97	95.7 / 48.7	1.97
5%	6513.1 / 3202.2	2.03	1319.2 / 658.0	2.02	359.5 / 231.2	1.53	178.3 / 98.98	1.80	82.3 / 48.4	1.70
10%	6629.1 / 33226	2.00	1332.3 / 6768	1.97	453.1 / 2435	1.86	186 / 1080	1.71	87.9 / 57.5	1.53
20%	8558.5 / 52522	1.64	1737 / 10733	1.62	561.3 / 384.5	1.46	226.3 / 1690	1.39	104.7 / 86.2	1.21
80%	97794 / 101654	.962	18658 / 22208	.836	4997 / 7528	.66	1487.6 / 3128	.48	501 / 1507	.33
ARITHMETIC AVERAGED α		1.86		1.72		1.80		1.46		1.35

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 352 A TEST LOCATION FPRC-OSU DATE 2-23-76

TEST FLOW 25 GPM BUBBLE POINT NA INITIAL CLEANLINESS 4.0/ml >12μm

TERMINAL ΔP 40 HOUSING ΔP 9.1 CLEAN ASSEMBLY ΔP 11.4 CLEAN ELEMENT ΔP 2.3

NET Δ P (37.7)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	12.3	13.3	15.2	18.9	26.5	41.6	49.1
Time (min.)	23.8	39.7	69.3	112.5	207.5	415.6	503.1

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	0.247	0.251	0.249
Gravimetric Level (mg/litre)	10499	9365	9932

BASE UPSTREAM LEVEL: 26.1 mg/litre

FINAL GRAVIMETRIC LEVEL: 249.0 mg/litre

ACCTD CAPACITY: 1244 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
UP / DOWN 2.5%	9307 / 6313	1.47	1409 / 782	1.80	383 / 159	2.41	138 / 38.8	3.56	51.9 / 13.9	3.73
UP / DOWN 5.0%	10493 / 7471	1.41	1539 / 959	1.61	409 / 239	1.71	140 / 72.9	1.92	52.5 / 25.8	2.04
UP / DOWN 10.0%	11281 / 7313	1.54	1744 / 893	1.95	488 / 204	2.39	166 / 61.4	2.70	60.9 / 22.7	2.68
UP / DOWN 20.0%	27560 / 22934	1.20	4107 / 3144	1.31	1053 / 826	1.28	349 / 301	1.16	131 / 123	1.07
UP / DOWN 80.0%	406803 / 405651	1.00	95220 / 93872	1.01	31486 / 29764	1.06	11741 / 10555	1.11	4729 / 3857	1.23
ARITHMETIC AVERAGED α		1.32		1.54		1.77		2.09		2.15

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 384 B TEST LOCATION FPAC-OSU DATE 3-6-75

TEST FLOW 40 GPM BUBBLE POINT 0.5" H_2O Seam Leak INITIAL CLEANLINESS 13/ml > 10μm

TERMINAL ΔP 40 psid HOUSING ΔP 19.2 CLEAN ASSEMBLY ΔP 19.9 CLEAN ELEMENT ΔP 0.7

NET Δ P (37.3)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	20.9	21.9	23.8	27.8	35.6	57.3	59.2
Time (min.)	45.5	57.8	64.7	69.2	73.1	76.6	77.7

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.250	.258	.254
Gravimetric Level (mg/litre)	17009	13461	15235

BASE UPSTREAM LEVEL: 25.6 mg/litre

FINAL GRAVIMETRIC LEVEL: 109 mg/litre

ACCTD CAPACITY: 300.7 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE		> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5%	UP	9568.8	2.29	667.9	124	210.8	153	83.84	233	39.64	198
	DOWN	4181.1		5.4		1.32		0.36		0.20	
5.0%	UP	9068.2	2.10	659.3	115	197.3	224	73.68	263	30.24	189
	DOWN	4326.1		5.72		0.88		0.28		0.16	
10.0%	UP	10299.3	2.02	662.2	70.8	190.4	154	69.48	116	28.52	89.1
	DOWN	5100.8		9.36		1.24		0.60		0.32	
20.0%	UP	11429.4	1.80	655.1	37.0	185.8	63.6	67.8	65.2	27.68	40.7
	DOWN	6362.2		17.7		2.92		1.04		0.68	
80.0%	UP	17580.6	1.27	764.6	16.1	212.1	76.9	77.96	62.9	31.44	39.3
	DOWN	13812.2		47.4		2.76		1.24		0.80	
ARITHMETIC AVERAGED α			1.90		72.6		134		148		111

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 384 C TEST LOCATION FPRC-OSU DATE 3-7-75

TEST FLOW 40gpm BUBBLE POINT 0.5" H_2O Seam Leak INITIAL CLEANLINESS 12/ml > 10μm

TERMINAL ΔP 40psid HOUSING ΔP 19.2 CLEAN ASSEMBLY ΔP 19.5 CLEAN ELEMENT ΔP 0.3

NET Δ P (39.7)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	20.5	21.5	23.5	27.4	35.4	57.3	59.2
Time (min.)	43.0	57.5	64.0	68.0	71.5	74.2	75.5

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	0.250	0.248	0.249
Gravimetric Level (mg/litre)	15811	14328	15069.5

BASE UPSTREAM LEVEL: 28.5 mg/litre

FINAL GRAVIMETRIC LEVEL: 82.6 mg/litre

ACCTD CAPACITY: 283.3 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE		> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5%	UP	4749.2	2.93	536.9	75.0	171.6	66.0	63.8	53.2	27.2	32.8
	DOWN	1622.4		7.16		2.6		1.2		0.72	
5.0%	UP	5397.2	3.16	593.8	151	181.6	157	69.1	144	29.3	122
	DOWN	1706.4		3.92		1.16		0.48		0.24	
10.0%	UP	5477.1	2.98	599.0	40.1	184.7	49.1	67.3	52.6	29.7	61.9
	DOWN	1838.0		14.9		3.8		1.28		0.48	
20.0%	UP	6703.6	2.47	641.3	35.0	195.6	53.7	72.6	67.2	31.8	66.3
	DOWN	2709.5		18.3		3.64		1.08		0.48	
80.0%	UP	8996.2	1.86	614.4	18.8	166.8	81.8	54.6	85.3	22.4	62.2
	DOWN	4836		32.7		2.04		0.64		0.36	
ARITHMETIC AVERAGED α			2.68		69.0		81.6		80.5		70.0

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 385A TEST LOCATION FPRC-OSU DATE 3-10-75

TEST FLOW 40gpm BUBBLE POINT 0.5"H_2O acet. INITIAL CLEANLINESS 9.6/ml > 10 μm

TERMINAL ΔP 40PSID HOUSING ΔP 19.2 CLEAN ASSEMBLY ΔP 19.3 CLEAN ELEMENT ΔP 0.1

NET Δ P (38.9)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	20.3	21.3	23.3	27.3	35.3	51.2	59.2
Time (min.)	87.5	95	99	102	104	106	107

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.250	.250	.248
Gravimetric Level (mg/litre)	11400	11180	11290

BASE UPSTREAM LEVEL: 18.49 mg/litre

FINAL GRAVIMETRIC LEVEL: 124 mg/litre

ACCTD CAPACITY: 299.6g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10µM	α_{10}	> 20µM	α_{20}	> 30µM	α_{30}	> 40µM	α_{40}	> 50µM	α_{50}
2.5%	30479.4 / 24936.2	1.22	866.1 / 323.8	2.68	190.6 / 16.96	11.2	78.4 / 1.28	61.2	38.72 / 0.64	60.5
5%	27478.6 / 22075	1.24	904.6 / 443.4	2.04	173.6 / 23.68	7.33	67.04 / 2.24	29.9	33.28 / 0.48	69.3
10%	36588.2 / 30583.8	1.20	1515.8 / 977.4	1.55	223.5 / 63.84	3.50	68.96 / 2.88	23.9	29.12 / 0.64	45.5
20%	48122.9 / 44038.2	1.09	2458.9 / 1952	1.26	311.36 / 148.8	2.09	79.68 / 6.08	13.1	30.88 / 0.16	193
80%	25918.5 / 19895.0	1.30	1739.7 / 1183.2	1.47	264.5 / 116.16	2.28	74.88 / 6.72	11.1	33.44 / 1.28	26.1
ARITHMETIC AVERAGED α		1.21		1.8		5.28		27.8		78.9

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 386A TEST LOCATION FPRC-OSU DATE 3-2-75

TEST FLOW 40gpm BUBBLE POINT .5 in H_2O acetam INITIAL CLEANLINESS 2.8 partk/ml > 10 μm

TERMINAL ΔP 40PSID HOUSING ΔP 19.2 CLEAN ASSEMBLY ΔP 21.3 CLEAN ELEMENT ΔP 2.1

NET Δ P (37.9)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	22.2	23.2	25.1	28.9	36.5	51.6	59.2
Time (min.)	24.0	31.0	39.0	47.5	55.0	60.0	61.2

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.250	.254	.252
Gravimetric Level (mg/litre)	11660	11070	11365

BASE UPSTREAM LEVEL: 18.9 mg/litre

FINAL GRAVIMETRIC LEVEL: 58 mg/litre

ACCTD CAPACITY: 175.3g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10µM	α_{10}	> 20µM	α_{20}	> 30µM	α_{30}	> 40µM	α_{40}	> 50µM	α_{50}
2.5%	3126.1 / 177.52	17.6	489.7 / 2.92	168	160.2 / .42	381	57.4 / .20	287	23.6 / .04	590
5%	2699.3 / 134.9	20.0	472.9 / 2.34	202	144.6 / .3	482	58.8 / .14	420	25.4 / .06	423
10%	2666.3 / 130.3	20.5	460.0 / 2.64	174	140.6 / 0.4	352	56.3 / .16	352	23.2 / .08	290
20%	2679.6 / 133	20.1	477.6 / 3.3	145	145.2 / .32	455	56.16 / .14	401	23.7 / .08	296
80%	2520.6 / 114.8	22.0	446.36 / 5.62	79.4	133.3 / 1.66	80.3	50.96 / .74	68.9	21.6 / .32	67.5
ARITHMETIC AVERAGED α		20.0		154		350		306		333

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 401A TEST LOCATION FPRC-OSU DATE 1-21-76

TEST FLOW 5 GPM BUBBLE POINT N/A INITIAL CLEANLINESS 4.3 /Ml >10μM

TERMINAL ΔP 40 HOUSING ΔP 3.6 CLEAN ASSEMBLY ΔP 5.6 CLEAN ELEMENT ΔP 2.0

NET Δ P (38)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	6.6	7.5	9.4	13.2	20.8	36.0	43.6
Time (min.)	61.7	65.1	67.5	70.6	73.9	*	*

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	0.250	0.250	0.250
Gravimetric Level (mg/litre)	2152	1876	2014

BASE UPSTREAM LEVEL: 26.6 mg/litre

FINAL GRAVIMETRIC LEVEL: 51.0 mg/litre

ACCTD CAPACITY: 37.2+ g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE		> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5%	UP	7991	2.12	683	14.1	197	123	75.8	190	34.4	∞
	DOWN	3776		48.4		1.6		0.4		0.0	
5.0%	UP	9296	1.65	715	4.97	164	34.2	58.4	73.0	24.0	120
	DOWN	5619		144		4.8		0.8		0.2	
10.0%	UP	10835	1.55	849	4.18	190	27.9	65.6	82.0	24.4	48.8
	DOWN	6971		203		6.8		0.8		0.5	
20.0%	UP	12360	1.42	921	3.05	182	19.0	59.8	59.8	23.8	119
	DOWN	8734		302		9.6		1.0		0.2	
80.0%	UP	—		—		—		—		—	
	DOWN										
ARITHMETIC AVERAGED α			1.69		6.58		51.1		101		∞

* RELIEF VALVE IN SPIN-ON CAN OPENED

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 402C TEST LOCATION FPRC-OSU DATE 2-6-76

TEST FLOW 5 GPM BUBBLE POINT N/A INITIAL CLEANLINESS 13.0/Ml >10μM

TERMINAL ΔP 10 HOUSING ΔP 3.9 CLEAN ASSEMBLY ΔP 5.2 CLEAN ELEMENT ΔP 1.3

NET Δ P (8.7)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	5.4	5.6	6.1	6.9	8.7	12.2	13.9
Time (min.)	8.4	15.7	24.9	40.1	48.7	50.4	51.2

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	0.248	0.245	0.247
Gravimetric Level (mg/litre)	1924	1860	1892

BASE UPSTREAM LEVEL: 24.6 mg/litre

FINAL GRAVIMETRIC LEVEL: 112 mg/litre

ACCTD CAPACITY: 23.9 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE		> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5%	UP	7291	1.42	596	5.2	143	28.6	50.1	41.8	18.9	36.4
	DOWN	5129		114		5.0		1.2		0.52	
5.0%	UP	10063	1.32	593	4.2	124	30.2	38.9	81.0	14.3	71.5
	DOWN	7642		141		4.1		0.48		0.20	
10.0%	UP	13605	1.28	704	5.1	163	18.7	56.9	74.9	22.6	565
	DOWN	10602		137		8.7		0.76		0.04	
20.0%	UP	11070	1.42	705	4.2	159	13.4	53.5	16.7	19.8	26.1
	DOWN	7800		167		11.9		3.2		0.76	
80.0%	UP	15542	1.18	1053	1.78	162	3.0	46.8	3.3	16.7	2.5
	DOWN	13212		592		54.2		14.2		6.6	
ARITHMETIC AVERAGED α			1.32		4.11		18.8		43.5		140

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 403 E TEST LOCATION FPRC-OSU DATE 1-22-76

TEST FLOW 5 GPM BUBBLE POINT N/A INITIAL CLEANLINESS 3.2/ml>10μM

TERMINAL ΔP 8 PSID HOUSING ΔP 3.6 CLEAN ASSEMBLY ΔP 6.4 CLEAN ELEMENT ΔP 2.8

NET Δ P (5.2)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	6.5	6.7	6.9	7.4	8.5	10.6	11.6
Time (min.)	6.5	7.8	8.7	10.2	16.4	20.1	25.3

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.250	.250	.250
Gravimetric Level (mg/litre)	1888	1722	1805

BASE UPSTREAM LEVEL: 23.8 mg/litre

FINAL GRAVIMETRIC LEVEL: 62.0 mg/litre

ACCTD CAPACITY: 11.4 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5% UP	13732	1.56	1569	1.71	316	1.93	84.9	2.27	27.1	2.98
2.5% DOWN	8819		920		164		37.4		9.1	
5.0% UP	14404	1.62	1610	2.05	347	2.13	105	3.28	36.2	5.84
5.0% DOWN	8893		784		163		32.0		6.2	
10.0% UP	14679	1.44	1551	1.74	340	2.30	106	3.40	38.6	4.95
10.0% DOWN	10185		890		148		31.2		7.8	
20.0% UP	14592	1.31	1454	1.83	300	2.40	84.9	3.30	28.0	5.00
20.0% DOWN	11112		794		125		25.7		5.6	
80.0% UP	7776	1.83	812	3.74	208	6.12	70.0	8.86	26.9	12.8
80.0% DOWN	4255		217		34.0		7.9		2.1	
ARITHMETIC AVERAGED α		1.55		2.21		2.97		4.22		6.31

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 404 A TEST LOCATION FPRC - OSU DATE 2-12-76

TEST FLOW 5 GPM BUBBLE POINT 1" H_2O INITIAL CLEANLINESS 9.0/ml > 10μm

TERMINAL ΔP 40.0 HOUSING ΔP 1.1 CLEAN ASSEMBLY ΔP 1.4 CLEAN ELEMENT ΔP .3

NET Δ P (39.7)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	2.4	3.4	5.4	9.3	17.3	33.2	41.1
Time (min.)	64.2	100.5	120.3	130.7	139.7	152.2	159.1

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	0.247	0.245	0.246
Gravimetric Level (mg/litre)	1975	1622	1798.5

BASE UPSTREAM LEVEL: 23.4 mg/litre

FINAL GRAVIMETRIC LEVEL: 27.0 mg/litre

ACCTD CAPACITY: 70.4 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5% UP	17225	1.63	1003	3.89	178	17.80	47.8	59.75	14.2	88.75
2.5% DOWN	10540		258		10.0		0.8		0.16	
5.0% UP	11981	1.55	798	3.31	139	14.33	31.8	56.79	8.2	102.5
5.0% DOWN	7752		241		9.7		0.56		0.08	
10.0% UP	12983	1.41	1010	2.32	174	8.02	48.3	100.63	16.0	∞
10.0% DOWN	9216		435		21.7		0.48		0.00	
20.0% UP	11097	1.49	992	2.48	168	7.34	47.4	36.46	14.9	∞
20.0% DOWN	7439		400		22.9		1.3		0.00	
80.0% UP	2869	1.82	348	3.82	77.3	16.10	20.1	50.25	5.0	83.33
80.0% DOWN	1575		91		4.8		0.4		0.06	
ARITHMETIC AVERAGED α		1.58		3.16		12.72		60.78		∞

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 405 D TEST LOCATION FPRC-OSU DATE 2-12-76

TEST FLOW 5 GPM BUBBLE POINT 3" H_2O INITIAL CLEANLINESS 11.5/ml >10μm

TERMINAL ΔP 40.0 HOUSING ΔP 0.5 CLEAN ASSEMBLY ΔP 0.9 CLEAN ELEMENT ΔP 0.4

NET Δ P (39.6)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	1.0	2.0	4.0	8.1	16.2	32.4	40.5
Time (min.)	67.3	82.0	88.0	91.7	98.6	105.8	108.7

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	0.253	0.244	0.249
Gravimetric Level (mg/litre)	1817	1650	1733

BASE UPSTREAM LEVEL: 22.8 mg/litre

FINAL GRAVIMETRIC LEVEL: 51.0 mg/litre

ACCTD CAPACITY: 46.8 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5% UP / DOWN	19002 / 12711	1.49	1194 / 416	2.87	157 / 25.6	6.13	41.2 / 2.4	17.2	11.2 / 0.4	28.0
5.0% UP / DOWN	18590 / 15461	1.20	1044 / 594	1.76	148 / 27.2	5.44	38.0 / 0.8	47.5	13.2 / 0.0	∞
10.0% UP / DOWN	19660 / 14009	1.42	1689 / 883	1.91	357 / 64.0	5.58	58.4 / 2.8	20.9	21.2 / 0.4	53.0
20.0% UP / DOWN	16462 / 12336	1.33	1336 / 840	1.59	308 / 93.6	3.29	42.4 / 4.4	9.64	14.8 / 0.4	37.0
80.0% UP / DOWN	8331 / 4419	1.89	1036 / 383	2.70	204 / 41.6	4.90	54.8 / 2.0	27.4	19.6 / 0.4	49.0
ARITHMETIC AVERAGED α		1.47		2.17		5.07		24.5		∞

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 406 E TEST LOCATION FPRC-OSU DATE 2-13-76

TEST FLOW 5 GPM BUBBLE POINT 3" H_2O INITIAL CLEANLINESS 7.3/ml >10μm

TERMINAL ΔP 40.0 HOUSING ΔP 1.1 CLEAN ASSEMBLY ΔP 1.3 CLEAN ELEMENT ΔP 0.2

NET Δ P (39.8)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	2.3	3.3	5.3	9.3	17.2	33.1	41.1
Time (min.)	96.1	103.8	107.9	113.8	125.5	146.1	155.4

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	0.250	0.252	0.251
Gravimetric Level (mg/litre)	1775	1660	1717.5

BASE UPSTREAM LEVEL: 22.8 mg/litre

FINAL GRAVIMETRIC LEVEL: 27.0 mg/litre

ACCTD CAPACITY: 67.0 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5% UP / DOWN	14648 / 9930	1.48	1239 / 465	2.67	216 / 27.6	7.83	66.0 / 2.2	30.0	25.2 / 0.24	105.0
5.0% UP / DOWN	13372 / 9296	1.44	1220 / 650	1.88	193 / 49.8	3.88	55.9 / 4.0	13.98	23.1 / 0.24	96.7
10.0% UP / DOWN	9295 / 5019	1.85	1114 / 371	3.0	336 / 31.6	10.63	676 / 3.1	21.81	23.4 / 0.4	58.5
20.0% UP / DOWN	6697 / 3176	2.11	886 / 273	3.25	354 / 26.8	13.21	53.9 / 1.8	29.94	19.3 / 0.08	241.7
80.0% UP / DOWN	4091 / 699	5.85	723 / 56.9	12.71	382 / 6.0	63.67	73.6 / 0.52	141.54	25.8 / 0.16	161.25
ARITHMETIC AVERAGED α		2.55		4.7		19.84		47.45		132.4

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 407 E TEST LOCATION FPRC-OSU DATE 2-16-76

TEST FLOW 5 GPM BUBBLE POINT 2" H_2O INITIAL CLEANLINESS 5.6/ml > 10μm

TERMINAL ΔP 40.0 HOUSING ΔP 0.9 CLEAN ASSEMBLY ΔP 1.1 CLEAN ELEMENT ΔP 0.2

NET Δ P (39.8)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	2.1	3.1	5.1	9.1	17.0	32.9	40.9
Time (min.)	81.3	88.7	93.0	100.3	114.0	143.1	158.7

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	0.248	0.246	0.247
Gravimetric Level (mg/litre)	1887	1662	1774

BASE UPSTREAM LEVEL: 23.2 mg/litre

FINAL GRAVIMETRIC LEVEL: 25.0 mg/litre

ACCTD CAPACITY: 69.6 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5% UP / DOWN	12258 / 8639	1.42	933 / 334	2.79	187 / 23.7	7.89	45.6 / 3.4	13.41	20.6 / 0.84	24.52
5.0% UP / DOWN	11592 / 7041	1.65	1097 / 400	2.74	174 / 35.6	4.89	54.9 / 5.2	10.56	17.6 / 1.8	9.78
10.0% UP / DOWN	10150 / 5393	1.88	1189 / 426	2.79	324 / 55.9	5.8	72.7 / 9.2	7.9	23.6 / 3.2	7.38
20.0% UP / DOWN	7023 / 3021	2.32	983 / 316	3.11	238 / 100	2.38	75.6 / 13.4	5.64	26.0 / 4.3	6.05
80.0% UP / DOWN	4080 / 1899	2.15	749 / 320	2.34	272 / 88.9	3.06	71.9 / 29.7	2.42	26.2 / 12.0	2.18
ARITHMETIC AVERAGED α		2.39		2.75		4.8		7.99		9.98

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 408 E TEST LOCATION FPRC-OSU DATE 2-16-76

TEST FLOW 5 GPM BUBBLE POINT 2½" H_2O INITIAL CLEANLINESS 7.9/ml > 10μm

TERMINAL ΔP 40 HOUSING ΔP 1.1 CLEAN ASSEMBLY ΔP 1.3 CLEAN ELEMENT ΔP 0.2

NET Δ P (39.8)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	2.3	3.3	5.3	9.3	17.2	33.1	41.1
Time (min.)	77.6	93.2	108.0	120.9	135.2	155.4	164.5

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	0.252	0.252	0.252
Gravimetric Level (mg/litre)	2047	1825	1936

BASE UPSTREAM LEVEL 25.8 mg/litre

FINAL GRAVIMETRIC LEVEL 16.0 mg/litre

ACCTD CAPACITY 80.3 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5% UP / DOWN	17295 / 12939	1.34	1052 / 409	2.57	186 / 22.2	8.38	52.1 / 1.4	37.21	20.5 / 0.32	64.06
5.0% UP / DOWN	14670 / 10123	1.45	1045 / 495	2.11	161 / 30.3	5.31	46.2 / 1.8	25.67	17.8 / 0.00	∞
10.0% UP / DOWN	9582 / 5648	1.7	965 / 324	2.98	199 / 22.1	9.01	52.6 / 0.8	65.75	16.6 / 0.16	103.75
20.0% UP / DOWN	6884 / 2951	2.33	884 / 224	3.95	203 / 20.2	10.05	62.2 / 2.6	23.92	22.0 / 0.56	39.29
80.0% UP / DOWN	3855 / 699	5.52	658 / 46.7	14.09	180 / 11.7	15.39	61.8 / 0.88	70.23	22.0 / 0.32	68.75
ARITHMETIC AVERAGED α		2.47		5.14		9.63		44.56		∞

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 410A TEST LOCATION FPRC-OSU DATE 4-16-75

TEST FLOW 12 GPM BUBBLE POINT 2.5" H_2O INITIAL CLEANLINESS 4.1 %/ml >10μm

TERMINAL ΔP 40 HOUSING ΔP 4.4 CLEAN ASSEMBLY ΔP 4.5 CLEAN ELEMENT ΔP 0.1

NET Δ P (39.9)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	5.5	6.5	8.5	12.5	20.5	36.4	44.4
Time (min.)	93.0	96.6	98.3	99.6	102.1	111.7	116.1

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.254	.251	.253
Gravimetric Level (mg/litre)	4652	4530	4591

BASE UPSTREAM LEVEL: 25.6 mg/litre

FINAL GRAVIMETRIC LEVEL: 47 mg/litre

ACCTD CAPACITY: 135 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5%	29151.4 / 253423	1.15	1596.7 / 1125.3	1.42	228.4 / 100.1	2.28	54.5 / 12.8	4.26	16.9 / 1.9	8.89
5%	41067 / 38946.5	1.05	3354.6 / 3107.6	1.08	476.2 / 345.8	1.38	99.6 / 45.1	2.21	33.3 / 8.6	3.87
10%	49376.8 / 46174.9	1.07	5103.6 / 4661.9	1.09	711.6 / 592.5	1.20	130.6 / 80.4	1.62	39.1 / 13.1	2.98
20%	43722.2 / 39489.8	1.11	4769.6 / 4145.6	1.15	1011.1 / 567.4	1.78	130.7 / 79.8	1.64	35.3 / 12.3	2.87
80%	5000.9 / 2544.9	1.97	806.8 / 293.5	2.75	241.3 / 58.1	4.15	86.2 / 13.9	6.20	40.9 / 4.7	8.9
ARITHMETIC AVERAGED α		1.27		1.48		2.16		3.19		5.50

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 410B TEST LOCATION FPRC-OSU DATE 4-25-75

TEST FLOW 12GPM BUBBLE POINT 2.5" H_2O INITIAL CLEANLINESS 9.2 %/ml >10μm

TERMINAL ΔP 40 HOUSING ΔP 4.2 CLEAN ASSEMBLY ΔP 4.3 CLEAN ELEMENT ΔP 0.1

NET Δ P (39.9)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	5.3	6.3	8.3	12.3	20.3	36.2	44.2
Time (min.)	74.7	77.2	78.5	79.6	80.8	84.0	87.2

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.250	.252	.251
Gravimetric Level (mg/litre)	6124	4736	5430

BASE UPSTREAM LEVEL: 30 mg/litre

FINAL GRAVIMETRIC LEVEL: 39 mg/litre

ACCTD CAPACITY: 118.8 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5%	43911.4 / 40723.7	1.08	2198.4 / 1625.1	1.35	318.9 / 134.9	2.36	81.3 / 17.3	4.70	25.4 / 3.68	6.9
5%	58315.8 / 52869.3	1.10	4351.7 / 3630.2	1.20	604.6 / 377.3	1.60	137.3 / 45.0	3.05	50.4 / 7.52	6.70
10%	79265.1 / 58727.7	1.35	9051.0 / 5735.0	1.58	1307.7 / 741.0	1.76	231.2 / 101.6	2.27	55.5 / 16.2	3.43
20%	70069.3 / 72539.0	.97	6965.4 / 8038.2	.87	983.4 / 1086.9	.909	175.2 / 150.4	1.16	51.2 / 24.0	2.13
80%	10878.6 / 7294.4	1.49	1539.5 / 870.1	1.77	365.9 / 158.9	2.30	109.4 / 30.4	3.60	41.4 / 9.44	4.39
ARITHMETIC AVERAGED α		1.20		1.35		1.79		2.96		4.71

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 410C TEST LOCATION EPRC-050 DATE 4-28-75

TEST FLOW 12 GPM BUBBLE POINT 2.5" H_2O INITIAL CLEANLINESS 7.4%/ml >10μm

TERMINAL ΔP 40 HOUSING ΔP 4.2 CLEAN ASSEMBLY ΔP 4.3 CLEAN ELEMENT ΔP 0.1

NET Δ P (39.9)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	5.3	6.3	8.3	12.3	20.3	36.2	44.2
Time (min.)	85.0	88.3	90.0	91.2	92.8	99.2	103.2

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.247	.255	.251
Gravimetric Level (mg/litre)	4854	4508	4681

BASE UPSTREAM LEVEL: 25.9 mg/litre

FINAL GRAVIMETRIC LEVEL: 30 mg/litre

ACCTD CAPACITY: 121.3 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5%	36519.2 / 31716.5	1.15	2005.9 / 1454.2	1.38	295.4 / 128.6	2.30	79.2 / 15.5	5.11	29.6 / 1.6	18.5
5%	45477.6 / 41606.4	1.09	3488.5 / 3068.5	1.14	486.6 / 332.6	1.46	104.2 / 40.6	2.57	36.6 / 7.04	5.20
10%	61019.7 / 56647.0	1.08	5999.0 / 5823.5	1.03	843.2 / 784.5	1.08	147.5 / 101.4	1.45	39.4 / 17.6	2.24
20%	67765.9 / 57918.7	1.17	8126.1 / 6492.6	1.25	1250.4 / 896.5	1.39	229. / 119.4	1.92	59.8 / 17.8	3.36
80%	6346.4 / 3579.2	1.77	998.7 / 478.9	2.09	253.0 / 94.7	2.67	84.6 / 19.2	4.40	33.6 / 4.64	7.24
ARITHMETIC AVERAGED α		1.25		1.38		1.78		3.09		7.31

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 333 AA TEST LOCATION LAB B DATE 5-20-75

TEST FLOW 20 gpm BUBBLE POINT — INITIAL CLEANLINESS 45.8

TERMINAL ΔP 20.0 HOUSING ΔP 2.1 CLEAN ASSEMBLY ΔP 3.2 CLEAN ELEMENT ΔP 1.1

NET Δ P (18.9)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	3.7	4.1	5.1	7.0	10.8	19.3	22.1
Time (min.)	9.00	15.25	23.00	32.25	39.25	41.75	42.50

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)			0.471
Gravimetric Level (mg/litre)	4124	3361	3743

BASE UPSTREAM LEVEL: 23.3 mg/litre

FINAL GRAVIMETRIC LEVEL: 34.7 mg/litre

ACCTD CAPACITY: 74.91 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
UP / DOWN 2.5%	4070 / 1230	3.3	770 / 65	11.8	242 / 5.8	41.7	122 / 1.7	71.8	52.7 / 0.8	65.9
UP / DOWN 5.0%	3930 / 1330	2.95	699 / 60.0	11.7	217 / 2.5	86.8	106 / 1.7	62.4	43.3 / 0.8	54.1
UP / DOWN 10.0%	3830 / 1520	2.52	644 / 81.6	7.89	188 / 10.8	17.4	89.1 / 5.0	17.8	42.5 / 3.3	12.9
UP / DOWN 20.0%	4120 / 1620	2.54	648 / 78.3	8.21	174 / 6.7	25.9	65.8 / 0.8	82.3	32.5 / 0	∞
UP / DOWN 80.0%	4910 / 2500	1.96	763 / 140	5.45	203 / 3.3	615	71.6 / 1.7	42.1	35.0 / 0	∞
ARITHMETIC AVERAGED α		2.65		9.01		46.7		55.3		∞

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 333 BB TEST LOCATION LABC DATE 3 June 1975

TEST FLOW 20 GPM BUBBLE POINT 5.6" H_2O INITIAL CLEANLINESS 6.4 >10μ

TERMINAL ΔP 43.70 HOUSING ΔP 3.7 CLEAN ASSEMBLY ΔP 6.40 CLEAN ELEMENT ΔP 2.70

NET Δ P (37.30)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	7.33	8.27	10.13	13.86	21.32	36.24	43.70
Time (min.)	22.85	31.70	39.55	44.37	47.58	49.97	50.58

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	0.48	0.52	0.50
Gravimetric Level (mg/litre)	2716.0	2869.0	2792.5

BASE UPSTREAM LEVEL: 18.44 mg/litre

FINAL GRAVIMETRIC LEVEL: 37.82 mg/litre

ACCTD CAPACITY: 70.62 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5% UP	4,520.91	1.45	429.99	5.69	129.90	20.33	60.60	74.81	37.50	73.53
2.5% DOWN	3,121.80		75.51		6.39		0.81		0.51	
5% UP	4,695.81	1.50	459.21	6.47	143.31	29.86	65.91	66.58	39.51	131.70
5% DOWN	3,127.50		71.01		4.80		0.99		0.30	
10% UP	4,336.29	1.66	458.58	6.58	139.71	35.02	66.00	110.00	41.41	197.19
10% DOWN	2,616.69		69.69		3.99		0.60		0.21	
20% UP	4,246.20	1.74	472.59	6.60	136.89	26.84	65.79	95.35	39.00	76.47
20% DOWN	2,440.80		71.61		5.10		0.69		0.51	
80% UP	7,280.40	1.62	678.69	3.94	171.99	14.81	80.70	53.80	49.84	83.15
80% DOWN	4491.60		172.29		11.61		1.50		0.60	
ARITHMETIC AVERAGED α		1.59		5.86		25.37		80.11		112.41

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 333CC TEST LOCATION LABC DATE June 3, 1975

TEST FLOW 20 GPM BUBBLE POINT 5.8" H_2O INITIAL CLEANLINESS 9.5 >10μ

TERMINAL ΔP 43.70 HOUSING ΔP 3.70 CLEAN ASSEMBLY ΔP 6.60 CLEAN ELEMENT ΔP 2.90

NET Δ P (37.10)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P	7.53	8.46	10.31	14.02	21.44	36.28	43.70
Time (min.)	21.88	27.00	34.37	38.38	41.32	43.45	44.03

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.53	.54	.54
Gravimetric Level (mg/litre)	3282.0	2621	2951.5

BASE UPSTREAM LEVEL: 21.05 mg/litre

FINAL GRAVIMETRIC LEVEL: 63.66 mg/litre

ACCTD CAPACITY: 70.18 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
2.5% UP	5,073.72	1.69	535.08	6.96	165.60	28.95	74.92	50.62	42.40	46.09
2.5% DOWN	3,002.80		76.92		5.72		1.48		0.92	
5% UP	4,908.40	1.74	531.32	7.41	164.40	34.25	71.88	54.45	41.88	104.70
5% DOWN	2,818.80		71.72		4.80		1.32		0.40	
10% UP	4,368.00	1.78	483.08	7.40	150.28	62.62	64.68	161.70	37.32	71.77
10% DOWN	2447.60		65.32		2.40		0.40		0.52	
20% UP	4,295.08	1.68	481.32	5.96	149.20	23.76	68.52	131.77	41.60	104.00
20% DOWN	2552.68		80.80		6.28		0.52		0.40	
80% UP	6218.12	1.30	601.72	3.36	164.68	15.08	70.00	58.33	44.12	84.85
80% DOWN	4798.28		179.08		10.92		1.20		0.52	
ARITHMETIC AVERAGED α		1.64		6.22		32.93		91.37		82.28

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 410 AA TEST LOCATION LABC DATE June 4, 1975

TEST FLOW 12 GPM BUBBLE POINT 3.4" H_2O INITIAL CLEANLINESS 4.5 >10μ

TERMINAL ΔP 44.8 HOUSING ΔP 4.8 CLEAN ASSEMBLY ΔP 5.0 CLEAN ELEMENT ΔP 0.2

NET Δ P 34.8	25%	5%	10%	20%	40%	80%	100%
Assembly Δ P	6.00	6.99	8.98	12.96	20.92	36.84	44.80
Time (min.)	89.73	91.02	91.73	92.72	95.40	107.32	112.37

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	0.50	0.50	0.50
Gravimetric Level (mg/litre)	2281.0	1827.0	2054.0

BASE UPSTREAM LEVEL 22.62 mg/litre

FINAL GRAVIMETRIC LEVEL 29.41 mg/litre

ACCTD CAPACITY: 115.44 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE		> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
25%	UP	43,792.00	0.97	4,059.00	0.97	622.00	1.06	158.00	1.17	74.60	1.57
	DOWN	44964.00		4,182.00		585.40		135.40		47.40	
5%	UP	49,817.40	1.13	5,530.60	1.24	945.40	1.43	244.60	1.74	104.00	2.14
	DOWN	43,941.00		4,475.40		660.00		140.60		48.60	
10%	UP	36,159.40	1.34	4,140.60	1.45	777.40	1.65	230.60	1.82	110.00	2.50
	DOWN	26,919.60		2,856.60		470.60		126.60		44.00	
20%	UP	9588.30	1.35	1329.70	1.12	338.00	0.88	134.00	0.69	80.00	0.60
	DOWN	7,120.60		1191.40		382.60		195.40		132.60	
80%	UP	3950.00	2.58	641.70	3.55	207.00	5.73	9900	9.34	61.00	13.35
	DOWN	1531.13		180.97		36.10		10.60		4.57	
ARITHMETIC AVERAGED α			1.47		1.67		2.15		2.95		4.03

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER 410 BB TEST LOCATION LABC DATE June 6, 1975

TEST FLOW 12 GPM BUBBLE POINT 3.0" H_2O INITIAL CLEANLINESS 0.6 >10μ

TERMINAL ΔP 44.1 HOUSING ΔP 4.1 CLEAN ASSEMBLY ΔP 4.2 CLEAN ELEMENT ΔP 0.1

NET Δ P 39.90	25%	5%	10%	20%	40%	80%	100%
Assembly Δ P	5.20	6.20	8.19	12.18	20.16	36.12	44.10
Time (min.)	85.30	87.17	88.57	89.97	92.10	102.88	108.92

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)	.53	.52	.53
Gravimetric Level (mg/litre)	2222.0	1969.0	2095.5

BASE UPSTREAM LEVEL 24.46 mg/litre

FINAL GRAVIMETRIC LEVEL 30.44 mg/litre

ACCTD CAPACITY: 120.95 g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE		> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
25%	UP	38,420.00	1.06	2952.50	1.05	462.50	0.95	138.25	0.91	60.75	0.84
	DOWN	36,190.00		2801.75		488.25		152.50		72.50	
5%	UP	44,990.00	0.96	4,147.50	0.99	646.75	1.02	184.25	1.45	81.75	1.58
	DOWN	46,789.25		4194.25		632.50		126.75		51.75	
10%	UP	49,516.75	1.18	5,530.00	1.36	1,013.25	1.76	273.25	2.65	111.75	2.74
	DOWN	42,065.00		4,060.75		575.00		103.25		40.75	
20%	UP	35,102.50	1.22	3,958.25	1.26	745.75	1.30	240.75	1.76	104.25	1.87
	DOWN	28,788.29		3,137.50		575.75		136.75		55.75	
80%	UP	44267.5	1.06	6592.5	1.35	1825.0	1.49	775.0	3.49	307.5	4.29
	DOWN	41,885.00		4,876.75		917.50		221.75		86.75	
ARITHMETIC AVERAGED α			1.10		1.20		1.40		2.05		2.25

APPENDIX B

PROPOSED LUBE OIL FILTER MULTI-PASS TEST PROCEDURE

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REVISED (1 February 1976)
PROJECT REVIEW DRAFT
Document No. OSU-LF-1

MULTIPASS METHOD FOR EVALUATING THE PARTICLE SEPARATION PERFORMANCE OF A LUBE OIL FILTER ELEMENT

Record of Action

9 December 1973	Project Undertaken
15 February 1974	Industrial Questionnaire
20 March 1974	Draft No. 1
26 March 1974	End-Item Advisory Review
27 March 1974	Manufacturer's Task Force Review
15 November 1974	Test Program Completed
9 December 1974	Project Review Draft
1 February 1976	Revised Project Review Draft

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TABLE OF CONTENTS

MULTIPASS METHOD FOR EVALUATING THE PARTICLE SEPARATION

CHARACTERISTICS OF A LUBE OIL FILTER ELEMENT

1.0 **INTRODUCTION** A lube oil filter is expected to possess structural integrity, particle separation performance, and resistance to agglomerative plugging. Each of these evaluation factors can be assessed by conducting individual tests.

This procedure deals only with the aspect of particle separation performance. It was intentionally designed to isolate all other factors which could influence the results.

Since the true purpose of a lube oil filter is to capture and retain abrasive type particles which are potentially harmful to the system, *a filter above all must display adequate particle separation characteristics.*

An important facet of this multi-pass lube oil filter test procedure is the nature of the test results. These results have mathematical significance in that they can be used to describe and predict the influence of the filter in an actual system.

2.0 **SCOPE** To include a multi-pass particle separation test for lube oil filter elements. It is a procedure for determining the contaminant capacity, particulate removal, and pressure loading characteristics. AC Coarse Test Dust (ACCTD) is used as the test contaminant.

3.0 **PURPOSE** To provide a reproducible test procedure for appraising the particle separation performance of a lube oil filter element

4.0 **REFERENCES**

4.1 International Standard Rules for the Use of Units of the International System of Units and a Selection of the Decimal Multiples and Sub-Multiples of SI Units, ISO/R 1000-1969.

4.2 International Standard Graphical Symbols for Hydraulic and Pneumatic Equipment and Accessories for Fluid Power Transmission, ISO/R 1219-1970. Agrees with ANSI/Y32.10-1967.

4.3 International Standards Organization Standard Draft Proposal Method for Calibration of Liquid Automatic Particle Counters Using "AC" Fine Test Dust, ISO/TC 131/SC 6 (USA-9) 12. (ANSI B93.28-1973)

4.4 American National Standard Procedure for Qualifying and Controlling Cleaning Methods for Hydraulic Fluid Power Fluid Sample Containers, ISO/TC 131/SC 6 (USA-8) 9. (ANSI/B93.20-1972)

4.5 Society of Automotive Engineers Determination of Hydraulic Pressure Drop, SAE/ARP 24B-1968.

4.6 American National Standard Method for Extracting Fluid Samples from the Lines of an Operating Hydraulic Fluid Power System (for Particulate Contamination Analysis), ISO/TC 131/SC 6 (USA-2) 3. (ANSI/B93.19-1972)

4.7 Society of Automotive Engineers Procedure for the Determination of Particulate Contamination in Hydraulic Fluids by the Control Gravimetric Procedure, SAE/ARP 785-1963.

4.8 American National Standard Method of Determining the Fabrication Integrity of a Hydraulic Fluid Power Filter Element, ISO/DIS 2942. (ANSI/B93.22-1972)

5.0 TERMS AND DEFINITIONS

5.1 *Filtration Ratio* (α_μ) The ratio of the number of particles greater than a given size (μ) in the influent fluid to the number of particles greater than the same size (μ) in the effluent fluid.*

5.2 *Terminal Pressure Drop* The maximum pressure drop permitted across the filter element as specified by the manufacturer before a change is required.

5.3 *Net Pressure Drop* The difference between the terminal pressure and the pressure drop across a clean element.

5.4 *ACCTD Capacity* The actual weight (grams) of AC Coarse test contaminant injected into the filter test system before the terminal pressure drop is reached.

5.5 *Multi-Pass Test* A test which requires the recirculation of unaltered effluent fluid through the filter element.

6.0 UNITS The International System of Units (SI) is used in accordance with Ref. No. [1].

7.0 LETTER SYMBOLS (Letter symbols are used in accordance with Ref. No. [1].)

8.0 GRAPHIC SYMBOLS Graphic symbols used are in accordance with Ref. No. [2].

9.0 GENERAL PROCEDURE OUTLINE

9.1 Set up and maintain apparatus per section 11.

9.2 Run all tests per sections 12, 13, and 14.

9.3 Analyze data from sections 12, 13, and 14 per section 15.

9.4 Present data from sections 14 and 15 per section 16.

*The filtration ratio, α, results from a multi-pass filter test using ACCTD as the test contaminant.

10.0 MEASUREMENT ACCURACY Measure flow, pressure, and temperature parameters within 2% of the true value.

11.0 TEST EQUIPMENT

11.1 Use a suitable timer for measuring minutes and fractions of minutes.

11.2 Use automatic particle counter calibrated per Ref. No. [3] or any ISO-approved counting method.

11.3 Use AC Coarse Test Dust.

11.4 Use sample bottles containing less than 1.5 particles per millilitre per bottle volume greater than 10 micrometres as qualified per Ref. [4].

11.5 Use test fluid conforming to Mil-H-5606, NATO symbol H-515, or DTD585 B Hydraulic Fluid Specification.

11.6 Use a filter performance test circuit comprised of a *"filter test system"* and a *"contaminant injection system"* as typified in Fig. B-1.

11.6.1 The filter test system consists of:

11.6.1.1 A reservoir constructed with a conical bottom displaying an included angle of not more than 90° with the entering oil diffused below the fluid surface.

11.6.1.2 A hydraulic pump which is essentially insensitive to contaminant at the operating pressures. WARNING: Pumps exhibiting excessive flow pulses will cause erroneous results.

11.6.1.3 A system clean-up filter capable of providing an initial system contamination level of less than 15 particles per millilitre greater than 10 micrometres.

11.6.1.4 Pressure gauges, temperature indicator, and controller and flowmeter.

11.6.1.5 Pressure taps in accordance with Ref. No. [5].

11.6.1.6 A turbulent sampling means located upstream and downstream of the test filter. Sample per Ref. No. [6] or any other ISO-approved sampling method.

11.6.1.7 Interconnecting lines which insure that turbulent mixing conditions exist throughout the filter test system and that contaminant traps, silting areas, and combinations of cyclonic separation zones and quiescent chambers are avoided.

11.6.2 The contaminant injection system consists of:

11.6.2.1 A reservoir constructed with a conical bottom displaying an included angle of not more than 90° with the entering oil diffused below the fluid surface.

11.6.2.2 A system clean-up filter capable of providing an initial contamination level of less than 1000 particles per millilitre greater than 10 micrometres and a gravimetric level less than 2 percent of the calculated level at which the test is being conducted.

11.6.2.3 A hydraulic pump (centrifugal or other types which do not alter the contaminant particle size distribution).

11.6.2.4 A sampling means for the extraction of a small flow (injection flow) from a point in the contaminant injection system where active circulation of fluid exists. Samples per Ref. No. [6].

8.6.2.5 Interconnecting lines which insure that turbulent mixing conditions exist throughout the contaminant injection system and that contaminant traps, silting areas, and combinations of cyclonic separation zones and quiescent chambers are not present. In particular, turbulent mixing conditions (average fluid velocity of greater than 20 feet per second) must exist throughout the length of the line conducting the injection fluid.

11.7 Use membranes and associated laboratory equipment suitable for conducting the double membrane gravimetric method per Ref. No. [7].

12.0 TEST FACILITY VALIDATION PROCEDURE

12.1 Validation of filter test system.

12.1.1 Validate at the minimum flow that the filter test system will be operated. NOTE: Install a conduit in place of a filter housing during validation.

12.1.2 Adjust the total test system fluid volume to be numerically equal to three litres plus one-fourth the minimum volume flow per minute value.

12.1.3 Contaminate the system fluid to a calculated gravimetric level of 25 milligrams per litre using AC Coarse Test Dust.

12.1.4 Circulate the fluid in the test system for one hour, extracting fluid samples at 15, 30, 45, and 60 minutes.

12.1.5 Analyze the four fluid samples and record three cumulate particle counts at 10, 20, 30, 40, and 50 micrometres for each sample.

12.1.6 Accept the validation test only if:

12.1.6.1 The average of all four particle counts obtained for a given size from each sample does not deviate more than 10 percent from the average particle counts of that size from all samples.

12.1.6.2 The average of all particle counts per millilitre at 10 micrometres is not less than 2000 nor more than 3000.

12.1.6.3 The particle counts per millilitre at 40 micrometres are not less than 60 nor more than 80.

12.2 Validation of contaminant injection system.

12.2.1 Validate at the maximum gravimetric level, maximum injection circuit volume, and the minimum injection flow rate to be used; refer to clauses 13.2.2 and 13.2.3.

12.2.2 Add the required quantity of contaminant in slurry form to the injection system fluid and circulate for at least 15 minutes.

12.2.3 Extract fluid samples at the point where the injection fluid is discharged into the filter test system at four equal time intervals based upon the depletion rate of the system. (The injection flow should not be stopped throughout this test.) Analyze each sample gravimetrically per Ref. No. [7].

12.2.4 Accept the validation test only if the gravimetric level of each sample is within ± 10 percent of the average of the four samples and ± 10 percent of the known gravimetric value.

13.0 PRELIMINARY PREPARATION

13.1 Test filter assembly.

13.1.1 Insure that test fluid cannot by-pass the filter element to be evaluated.

13.1.2 Subject the test filter element to a fabrication integrity test in accordance with Ref. No. [8].

13.1.2.1 Disqualify the element from further testing if it fails to exhibit at least the designated test pressure.

13.1.2.2 Where applicable, allow the fluid to evaporate from the test filter element before installing in the test filter housing.

13.2 Contaminant injection system.

13.2.1 Using 25 mg/litre as a base upstream gravimetric level, calculate the predicted test time (τ') in minutes by the following equation:

$$\tau' = \frac{\text{(Estimated ACCTD Capacity of Filter Element, mg)}}{\text{(25 mg/litre) (Test Flow Rate, litres/minute)}}$$

A second element may be tested for capacity analysis if the estimated value of the ACCTD apparent capacity of the test element is not supplied by the filter manufacturer.

13.2.2 Calculate the minimum required operating injection system volume (σ, litres) which is compatible with the above predicted test time (τ') and a value for the injection flow of 0.5 litres/minute using the following equation:

$$\sigma = 1.2\ (\tau', \text{ minutes})\ (\text{Injection Flow, litres/minute})$$

NOTES:

1. The volume calculated above will assure a sufficient quantity of contaminated fluid to load the test element plus 20 percent for adequate circulation throughout the test. Larger injection system volumes may be used.

2. The 0.5 litres/minute value of the injection flow insures that the downstream sample flow expelled from the filter test system will not significantly influence the test results at the lower flow rate restriction given in the scope. Lower injection flow rates may be used provided that the desired base upstream gravimetric level is maintained. An injection flow rate below 0.25 litres/minute is not recommended due to silting characteristics and accuracy limitation.

13.2.3 Calculate the gravimetric level (γ', mg/litre) of the injection system fluid using the following equation:

$$\gamma' = \frac{\text{(25 mg/litre) (Test Flow, litres/minute)}}{\text{(Injection Flow, litres/minute)}}$$

13.2.4 Calculate the quantity of contaminant (ω, grams) needed for the contaminant injection system by the following equation:

$$\omega\text{(grams)} = \frac{(\gamma'\text{, mg/litre)(Injection System Volume, litres)}}{1000}$$

13.2.5 Adjust the injection flow rate at stabilized temperature to within $\pm$ 5 percent of the value selected in clause 13.2.2 and maintain throughout the test.

13.2.6 Adjust the total volume of the contaminant injection system to the value determined in clause 13.2.2.

13.2.7 Circulate the fluid in the contaminant injection system through its system clean-up filter until a contamination level of less than 1000 particles per millilitre greater than 10 micrometres and a gravimetric level less than 2 percent of the value determined in clause 13.2.3 are attained.

13.2.8 By-pass the system clean-up filter after the required initial contamination level has been achieved.

13.2.9 Add in slurry form the quantity (grams of contaminant determined in clause 13.2.4) to the injection system reservoir.

13.2.10 Circulate the fluid in the injection system for a minimum of 15 minutes to thoroughly disperse the contaminant.

13.3 Filter test system.

13.3.1 Install the filter housing (without the test element) in the filter test system.

13.3.2 Circulate the fluid in the filter test system at rated flow and a stabilized test temperature of 38° $\pm$ 2°C and record the pressure drop of the empty filter housing.

13.3.3 Adjust the total fluid volume of the filter test system (exclusive of the system clean-up filter circuit) such that it is numerically equal to three litres plus one-fourth the designated test volume flow through the filter per minute.

13.3.4 Circulate the fluid in the filter test system through the system clean-up filter until a contamination level of less than 15 particles per millilitre greater than 10 micrometres is attained.

13.3.5 Select and install the proper lengths of capillary tubing upstream and downstream of the test filter such that the initial upstream sample flow is approximately 0.6 times the injection flow and the downstream sample flow is within 5 percent of the injection flow. Maintain uninterrupted flow from the two sampling points during the entire test.

13.3.6 Return the sampling flow upstream of the test filter directly to the reservoir when sampling is not in progress.

13.3.7 Collect the sampling flow downstream of the test filter outside the filter test system in order to assist in maintaining a constant system volume which should be kept within 15 percent of the required system volume.

14.0 FILTER PERFORMANCE TEST

14.1 Install the filter element into its housing and subject the assembly to the specified test condition (test flow and test temperature of 38° $\pm$ 2°C)and reaffirm fluid level.

14.2 Measure and record the clean assembly pressure drop. Calculate and record the clean element pressure drop. Clean assembly minus the housing pressure drop measured in clause 13.3.2.

14.3 Calculate the pressure drops corresponding to increases of 2.5, 5, 10, 20, 40, 80, and 100 percent of the net pressure drop.

14.4 Obtain a sample from upstream of the test filter element to determine the initial system contamination level.

NOTE: Take all samples in such a manner so as to minimize the aeration of the fluid sample.

14.5 Obtain a sample from the contaminant injection system.

14.6 Measure and record the injection flow rate.

14.7 Initiate the filter test as follows:

14.7.1 By-pass the system clean-up filter.

14.7.2 Allow the injection flow to enter the filter test system reservoir.

14.7.3 Start the timer.

14.7.4 Start the downstream sample flow.

14.8 Extract upstream and downstream samples simultaneously when the pressure drop across the filter assembly has increased by 2.5 $\pm$ 1 percent of the net pressure drop. Record the test time to reach the 2.5 percent point.

NOTE: Use identical sample time of not more than 30 seconds for both upstream and downstream samples. Since the sampling procedure

requires the sample volume to be within 50 to 90 percent of the sample bottle volume, more than one size sample bottle may be required.

14.9 Repeat clause 14.8 at increases of 5 ± 1, 10 ± 1, 20 ± 2, 40 ± 2, and 80 ± 2 percent of the net pressure drop. No sample is required at the 40% point.

14.10 Record the test time (χ, minutes) for the pressure drop across the filter assembly to increase by 100 percent of the net pressure drop.

14.11 Conclude the test by stopping the flow to the test filter.

14.12 Obtain a fluid sample from the contaminant injection system.

14.13 Measure and record the injection flow rate.

15.0 SAMPLE EVALUATION

15.1 Analyze the samples extracted from the filter test system by determining the number of particles per millilitre greater than 10, 20, 30, 40, and 50 micrometres with an automatic particle counter calibrated per Ref. No. [3] or any ISO-approved counting method. Obtain a minimum of three particle counts for each fluid sample and calculate the arithmetic average for each size range counted.

NOTE: Care should be taken to dilute samples appropriately to avoid exceeding the saturation limit of the counting method determined by the approved calibration procedure.

15.2 Conduct a gravimetric analysis per Ref. No. [7] on the two samples extracted from the contaminant injection system and on the upstream sample extracted from the filter test system at the 80 percent sample point.

NOTE: The final sample is taken at the 80 percent point because it often overlaps the 100 percent point.

15.2.1 Record the 80 percent gravimetric value as the final system gravimetric level.

15.2.2 Calculate the average (γ) of the two gravimetric levels from the injection system.

15.2.3 Accept the test only if the gravimetric level of each sample is within 10 percent of the average from 15.2.2.

15.3 Calculate and record the injection flow rate by averaging the measurements taken at the beginning and end of the test. Accept the test only if this value is equal to the selected value $\pm$ 5 percent.

15.4 Calculate and record the actual base upstream gravimetric level by multiplying the average injection gravimetric level (γ, mg/litre) by the average injection flow rate (litres/ minutes) per clause 15.3 and dividing by the test flow (litres, minute). Accept the test only if this value is equal to 25 $\pm$ 2.5 mg/litre.

15.5 Calculate the filtration ratio for each particle size and sample point as defined in Section 5 and average the five values at each particle size. Record these calculations as shown in Fig. B-2.

16.0 DATA PRESENTATION

16.1 Report the following minimum information for filter elements evaluated using this recommended standard. Present all test and calculation results as shown in Fig. B-2.

16.2 Using the actual test time (τ) to reach the terminal pressure drop, the average gravimetric level (γ) of the injection stream, and the injection flow rate, calculate the filter element ACCTD Capacity using the following equation:

$$\text{ACCTD Capacity (grams)} = \frac{(\gamma,\ \text{mg/litre})(\text{Injection Flow Rate, litres/minute})(\tau,\ \text{minutes})}{1000}$$

Record the ACCTD Capacity as shown in Fig. B-2.

16.3 Report the values of the gravimetric levels obtained in Clause 15.2.

16.4 Plot the average filtration ratios versus particle size on log linear paper, as illustrated in Fig. B-3.

16.5 Have available a record of all the following minimum test data in all test reports referencing this recommended standard:

16.5.1 All physical values pertaining to the test.

16.5.2 All additional provisions or modifications pertaining to the test.

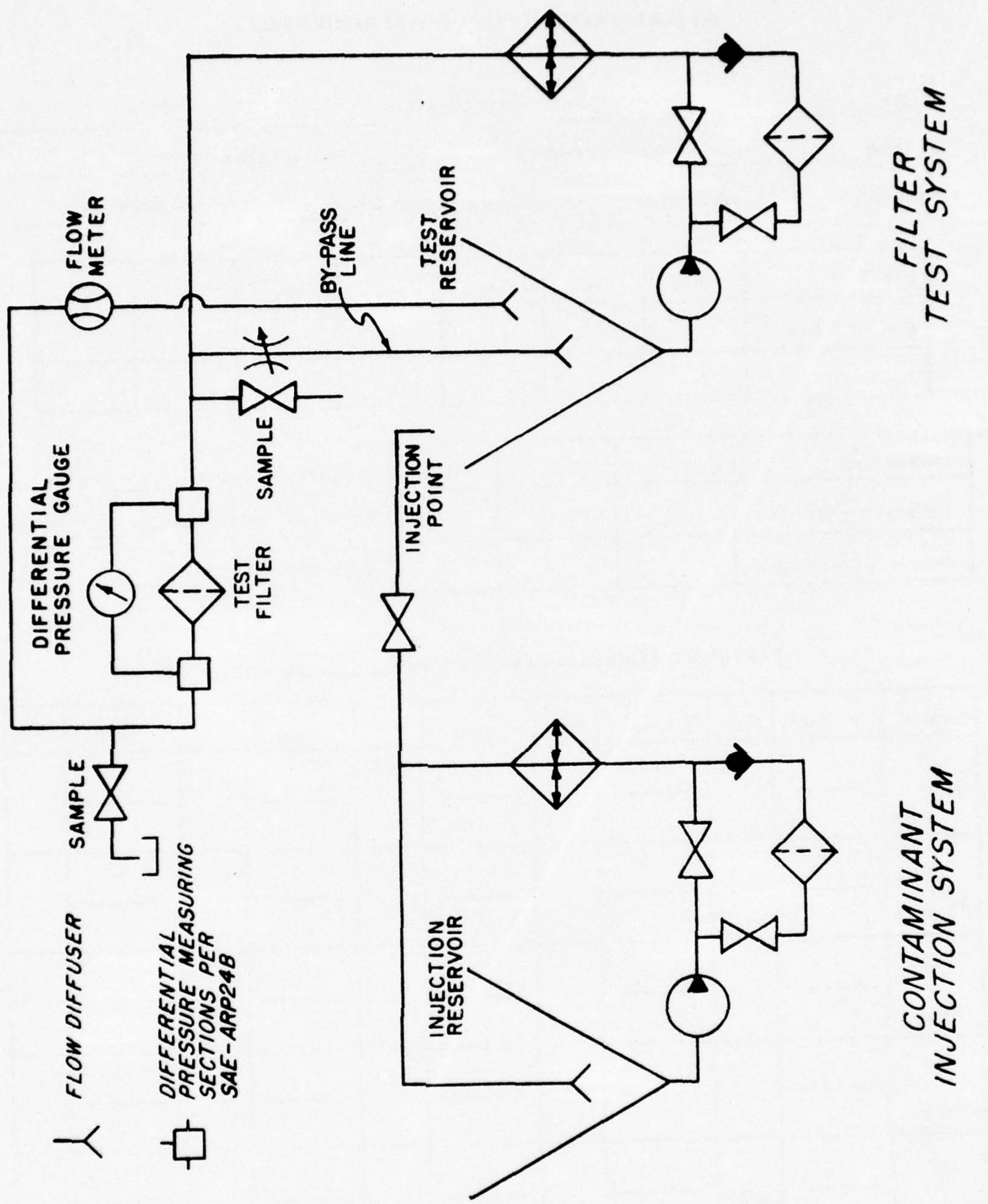

Fig. B-1. Typical Performance Test Circuit.

FILTER ELEMENT MULTI-PASS TEST REPORT SHEET

FILTER ________________ TEST LOCATION ____________ DATE ________

TEST FLOW ________ BUBBLE POINT ________ INITIAL CLEANLINESS ________

TERMINAL ΔP ____ HOUSING ΔP ____ CLEAN ASSEMBLY ΔP ____ CLEAN ELEMENT ΔP ____

NET Δ P (____)	2.5%	5%	10%	20%	40%	80%	100%
Assembly Δ P							
Time (min.)							

Injection Fluid	Initial	Final	Average
Injection Flow Rate (LPM)			
Gravimetric Level (mg/litre)			

BASE UPSTREAM LEVEL: ____ mg/litre

FINAL GRAVIMETRIC LEVEL: ____ mg/litre

ACCTD CAPACITY: ____ g

PARTICLE DISTRIBUTION ANALYSIS (PARTICLES PER MILLILITRE)

SAMPLE	> 10μM	α_{10}	> 20μM	α_{20}	> 30μM	α_{30}	> 40μM	α_{40}	> 50μM	α_{50}
UP 2.5% DOWN										
UP 5% DOWN										
UP 10% DOWN										
UP 20% DOWN										
UP 80% DOWN										
ARITHMETIC AVERAGED α										
MINIMUM α										

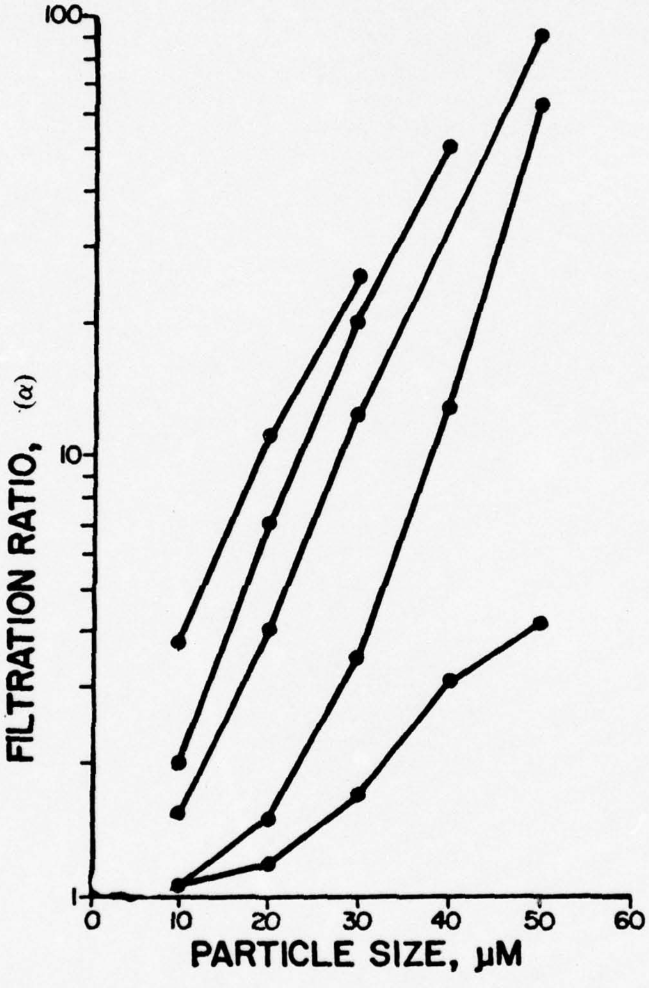

Fig. B-3. Typical Beta Graphs for Lube Oil Filters.

APPENDIX C

(PROPOSED)

COMPONENT PERFORMANCE SPECIFICATION

FILTER, LUBRICATING OIL, DISPOSABLE

OFF-HIGHWAY APPLICATION

2nd Draft
February 1976

(PROPOSED)

COMPONENT PERFORMANCE SPECIFICATION
FILTER, LUBRICATION OIL, DISPOSITION
OFF-HIGHWAY APPLICATION

NOTE: This draft was prepared by personnel of Oklahoma State University as part of a program with the U.S. Army Mobility Equipment Research and Development Center with the cooperation of various filter and engine manufacturers. This draft reflects the results of a test verification program which was conducted as well as current military requirements.

1.0 SCOPE

1.1 *Intended Use* This specification covers oil filters intended primarily for application to crankcase lubrication systems of engines installed in earthmoving equipment, motor trucks, and similar commercially built ground vehicles used primarily for off-highway purposes.

1.2 *Inclusion* This specification includes those aspects of a lubricating oil filter concerned with its performance capabilities, its mechanical strength, and its durability.

2.0 PURPOSE

2.1 *Requirements* This specification establishes the specific requirements of a filter element when tested in accordance with the designated procedure.

2.2 *Test Procedures* This specification requires the use of test procedures proposed by Oklahoma State University.

3.0 REQUIREMENTS

3.1 *Rated Conditions* The manufacturer shall specify the rated flow and minimum bubble point for all test elements submitted. Mil-L-2104 fluid is used in all tests except the particle separation test, which requires Mil-H-5606. When required, the terminal pressure drop shall be 40 psid.

3.2 *Performance* The filter shall meet the specific requirements specified herein.

3.2.1 *Particle Separation Characteristics* When tested in accordance with the multi-pass particle separation test, the filter element shall exhibit the following performance values:

1. **An average 10μM filtration ratio (α_{10}) equal to or greater than 2.0.**
2. **An ACCTD capacity of at least 1.5 grams per lpm of rated flow.**
3. **A clean element pressure drop of .50 bar differential (7.25 psid) or less.**

3.2.2 *Sludge Removal Characteristics* (Awaiting action by SAE Lube Oil Committee)

3.2.3 *Element Collapse* When tested in accordance with the collapse/burst resistance test, the filter shall not exhibit any decrease in the slope of the pressure drop versus grams added curve before a differential pressure of 7 bar differential (101.5 psid) is reached.

3.2.4 *Material Compatibility* When tested in accordance with the material compatibility test using a maximum test temperature of 135°C (275°F), the filter element shall not exhibit a decrease in the pressure drop vs. grams added curve before a differential pressure of 7 bar differential (101.5 psid) is reached.

3.2.5 *Media Migration* When tested in accordance with the media migration test, the average media migration per element shall not exceed 10 milligrams.

3.2.6 *Additive Removal* When tested in accordance with the Ash Type Oil Additive Removal Test, additive removal shall not exceed five percent (5%).

3.2.7 *Anti-Drainback Characteristics* When tested in accordance with the anti-drainback valve test, the average leakage shall not exceed one percent (1%) of the total fluid volume contained within the filter assembly per hour.

3.2.8 *Vibration Fatigue Resistance* When tested in accordance with the vibration fatigue test, the filter assembly shall last at least 72 hours without evidence of failure.

3.2.9 *Burst Pressure* When tested in accordance with the hydrostatic burst pressure test, the pressure at which the filter assembly bursts shall not be less than 14 bar differential (203 psid).

3.2.10 *Relief Valve Characteristics* When tested in accordance with the relief valve performance test, the leakage rate shall not exceed 1% of the rated flow of the filter at a pressure drop of 125% of the terminal pressure drop. The relief valve shall exhibit a pressure drop of less than 80% of the collapse pressure at the rated flow of the element.

SECTION V

ON-BOARD MONITOR STUDY

PROJECT STAFF

Gary A. Roberts, Program Manager

R. L. Decker, Project Engineer

R. L. Brown, Staff Engineer

M. T. Yokley, Project Associate

FOREWORD

The On-Board Monitor Study is a continuing data acquisition effort utilizing the Statistical Analog Monitor. The principle objectives of this year's effort were to obtain base line data from operating industrial equipment in the field, monitor tests of government equipment at MERDC, and continue the development of the necessary hardware. This report discusses the project activities and presents a summary of the data obtained.

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CHAPTER I

INTRODUCTION

The On-Board Monitor Project is a continuing data acquisition effort utilizing the Statistical Analog Monitor (STAM). The development of the STAM and the theoretical basis for the data acquisition method are documented in earlier reports. During 1974, a cooperative program was begun with 15 major companies which are producers of mobile equipment of the type utilized by the U.S. Army. The cooperating companies installed monitors on operating vehicles to obtain representative duty cycles. The objectives of this year's program were to provide the necessary monitors for various tests conducted by MERDC.

There were two major errors in the initial assumptions upon which the program was based. The program plan did not provide for installation support of the monitors. The installation requires only that conventional transducers be placed in the hydraulic system and a direct connection to the vehicle power supply. Previous tests at military installations had shown that the only significant problem was transducer location and the manufacturer should be well equipped to cope with this. Unfortunately, the administrative problems were overlooked. In many cases, several months elapsed before the units were installed. In some cases, the transducers were installed in the wrong points in the system. Severe damage resulted from improper installation. Units were mounted on exhaust manifolds, cables were severed by inspection plates, and transducers were destroyed when placed in locations exposed to external mechanical force (e.g., used as an operator step).

The second major fallacy was that the cooperating companies would accumulate test hours very rapidly. In fact, some of the first STAM units distributed had operated over 100 hours in the first two weeks after delivery. A severe recession in the industrial and construction machinery industry practically eliminated test programs. Those tests which were conducted

were of such severity that equipment down time exceeded operational time by as much as a factor of 100.

The extensive preliminary testing on vehicles also proved misleading in one technical area. The test vehicles were equipped with military specified electromagnetic noise suppression systems. These emissions from the commercial vehicles have proven extremely vexing, and overcoming this problem has consumed more project man hours than any other single area. It is interesting to note that the EMI problem affects conventional instrumentation just as much as it affects STAM. The problem is normally disregarded, and data which do not conform to the expected are ignored.

If the data obtained were disappointing in quantity, the success in terms of content was more than offsetting. Various types of hydraulic systems can be identified by the most casual observation of the STAM results. In several instances, serious system problems have been detected and corrected in preproduction tests. Several of the participating sponsors are committed to direct financial support of the program in the future. It is this commitment which will assure that the quantity and quality of data will improve rapidly.

This report is primarily a presentation of the data obtained. The hardware developments and field experience are discussed briefly together with a review of some of the more interesting results. The balance of the report consists of the STAM results, including equivalent duty cycle reconstructions.

CHAPTER II

HARDWARE DEVELOPMENTS

The major portion of the On-Board Monitor Project manpower and funds is expended on hardware development and support. During the past year, this effort has been divided into four general areas – maintenance of field units, development of specialized recording features (primarily for in-house use by MERDC), fabrication of the Type IV readout device, and developing solutions for the electromagnetic noise problem.

It is easy to lose perspective of the maintenance requirements of the operating units. The STAM was designed and tested for very severe use conditions and has routinely survived 500 hour excursions on operating vehicles. On the other hand, more than 100 units are now available with typically 50 in the field at any time. In the course of the testing program, over 75% of the vehicles have themselves failed during the time STAM monitors were installed. The monitor has demonstrated a mean time between failures of 1600 hours, an excellent record by any standard. Unfortunately, the survival rate of the cables and transducers is terrible, with nearly half of all packages supplied damaged by the user. Some of this is, of course, unavoidable due to the exposure imposed by the installation possibilities. The labor required to prepare a new set of cables and calibrate a new transducer exceeds that of constructing the monitor itself.

Apart from damage, the major source of loss of data has come from electromagnetic interference. Extensive testing has shown that STAM is relatively insensitive to EMI, except for very intense fields or very high frequencies. The earliest problems encountered were on gasoline powered forklifts and were attributed to ignition noise. The addition of the automatic end of test shutdown on the Mk. IIIC monitors brought the problem to an immediate urgency.

Noise which would affect the data also triggered the shutdown SCR. These false shutdowns triggered the operator light. In the most severe case, the unit would not remain on long enough to complete engine startup. On most vehicles, the false triggering would occur every 5 to 20 hours. The sensitivity of the SCR to any trigger input is a function of temperature; and, in general, STAM operates at a fairly high temperature.

The shutdown circuit was not designed to hold the unit off for long periods. The intent was to disconnect the monitor as soon as the warning light was illuminated. The incandescent bulb frequently failed, and the operator was not alerted. In addition, there was widespread complaint that there was no indication that the monitor was actually on under any conditions.

In order to solve these problems without performing a complete design iteration which would obsolete the existing monitors, an add-on unit was designed for the monitor. This end-of-test annunciator is connected to the main power line. The STAM power is supplied from the annunciator box, and a sense line from the STAM end-of-test detector circuit is returned. If the STAM is receiving power, a small light emitting diode on the annunciator is lighted. When the desired test time is reached, an electronic device (Sonalert) emits a loud tone. The monitor is not shut off, but the tone continues until power is removed. The test time is controlled by one board of the monitor, but both test timers are operating. Thus, the second board, which is set for a much longer time, will record the actual test time if the monitor is not disconnected. The annunciator also incorporates a power line filter which has been demonstrated effective in field tests on vehicles which exhibited noise problems. The complete assembly is shown in Fig. 2-1.

Apart from the shutdown circuit, the STAM laboratory experiments had shown virtually no susceptibility to power line noise. In field tests by project personnel in Arizona, it was observed that the very high frequency pulses generated by solenoid valves appeared on the power line and also on the pressure transducer signal. The temperature recorders were unaffected. This phenomenon results from the amplifier configuration and the

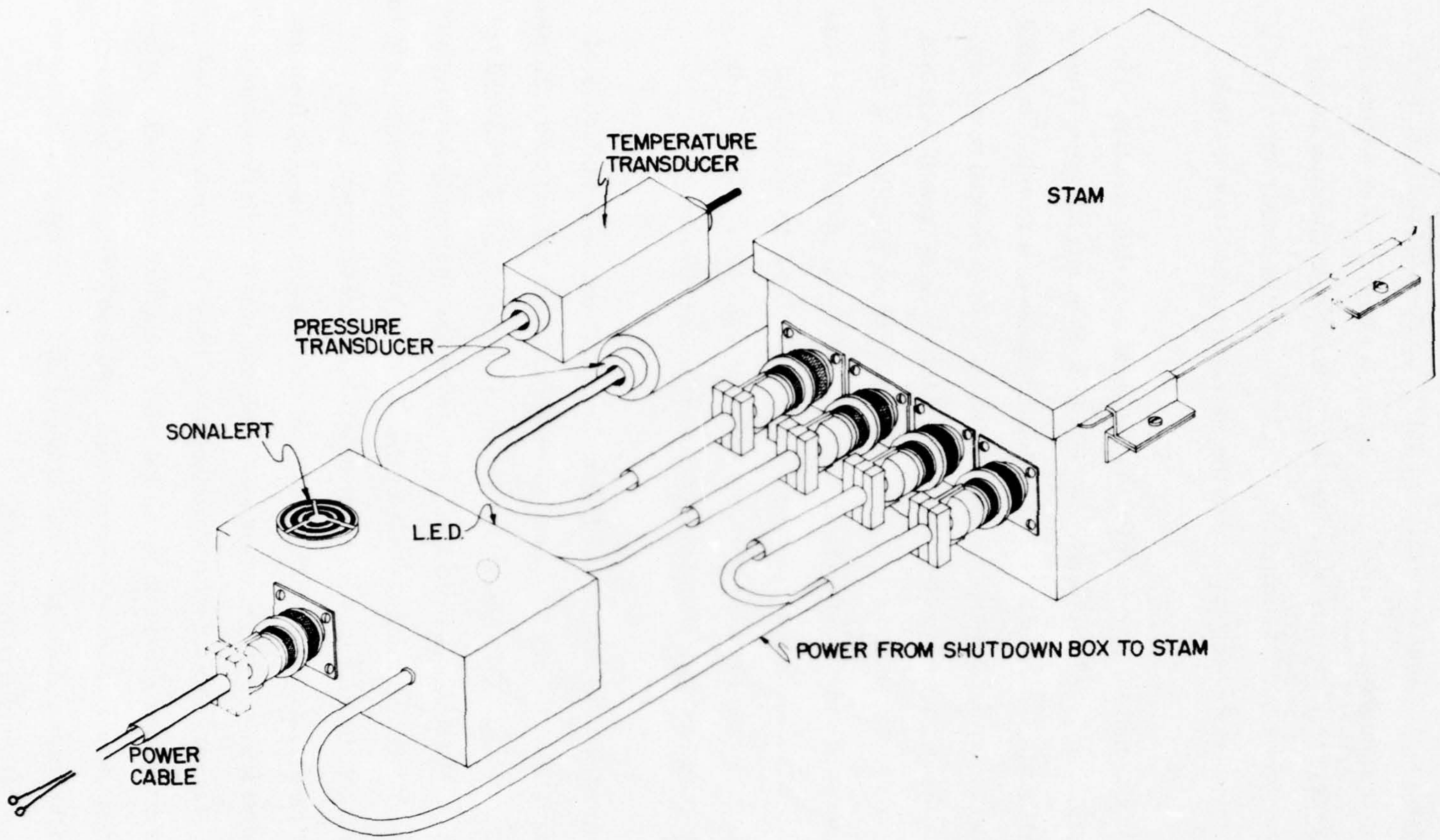

Fig. 2-1. STAM Assembly with End-of-Test Annunciator.

transducer characteristics. The amplifier used in STAM is a single-ended configuration in which noise received in common mode is not amplified but is passed through without attention. Further, the operational amplifier in use had very poor rejection characteristics of noise on the amplifier power lines. Substitution of a CMOS amplifier greatly improves this condition, and all amplifiers are being replaced as the units are returned from the field.

At any point in a circuit, the ratio of signal to noise is the important criterion. The output of the bonded strain gage transducers in use is only 40 millivolts full scale. Thus, a small noise imposed on the transducer sense line is amplified greatly. A recently developed integrated circuit transducer was obtained for testing. This device has internal power regulation and amplification and outputs a 10v signal. The high level output greatly improves the signal-to-noise ratio and, at the same time, eliminates the high amplification requirements. In addition, the transducer is field interchangeable, which reduces the calibration work load. Only one of these transducers was available until recently, and it has been installed and furnished to MERDC. Complete changeover to these devices is not economically feasible at present; but, as new transducers are procured, replacement will be effected.

One obstacle to efficient solution of the EMI problem has been the identification of noise sources. The presence of solenoid valves on mobile equipment had not been anticipated, and the experience of using STAM with a-c solenoid valves on hydraulic test stands had not revealed any problem. During the Arizona tests, it was found that the backup warning horn on many vehicles is air operated, using a solenoid valve. A very serious EMI source was revealed through the efforts of Mr. Russ Janke of International Harvester Company. Based on experience in real-time data acquisition, Janke suspected the hour meters used on most test vehicles. These meters are clockwork operated with a solenoid which resets the spring at frequent intervals. A meter furnished by IHC for laboratory tests consistently activated the STAM shutdown and recorded a count on the STAM event counters at every level. The noise suppression techniques developed for the solenoid valves were ineffective. At the present time, only the combined power line filter, CMOS amplifier, and high voltage transducer are sufficient to assure rejection of this interference.

One of the most interesting assignments presented to the project team was monitoring of the input power duty cycles of mobile equipment. Such information has considerable utility. Substantial fuel savings can be obtained by proper design and/or selection of the internal combustion engines. In some cases, the use of battery powered electrical drives can be advantageous.

The use of torque meters or other specialized instrumentation is not practical for a broad study. However, for a given engine, the power output can be closely approximated from the engine speed and manifold pressure. The gross horsepower available is a function of RPM, while the loading of the engine is reflected by the manifold pressure. The solution chosen was to compute the output horsepower of the engine and statistically monitor this computed variable.

A typical plot of gross horsepower vs. engine speed for a Hyster forklift is shown in Fig. 2-2. The engine vacuum vs. percent engine load is shown in Fig. 2-3. The operating horsepower is:

$$\text{Operating HP} \quad = \quad \text{(Gross HP) (\% Engine Load)}$$

Note that an approximate curve is shown on both of these plots. The use of these straight-line approximations facilitates the calculation of the operating horsepower.

Fig. 2-4 is a block diagram of the system which performs the required calculations. The engine speed is determined by converting the ignition system output to a D. C. signal. This signal is converted to gross horsepower by a nonlinear conversion. The gross horsepower and engine vacuum are multiplied to obtain operating horsepower.

Fig. 2-5 is the schematic of the circuit which was constructed to implement the horsepower calculation. The tach generator circuit is connected to the distributor points. This circuit generates a D. C. signal proportional to the engine speed. This D. C. signal is

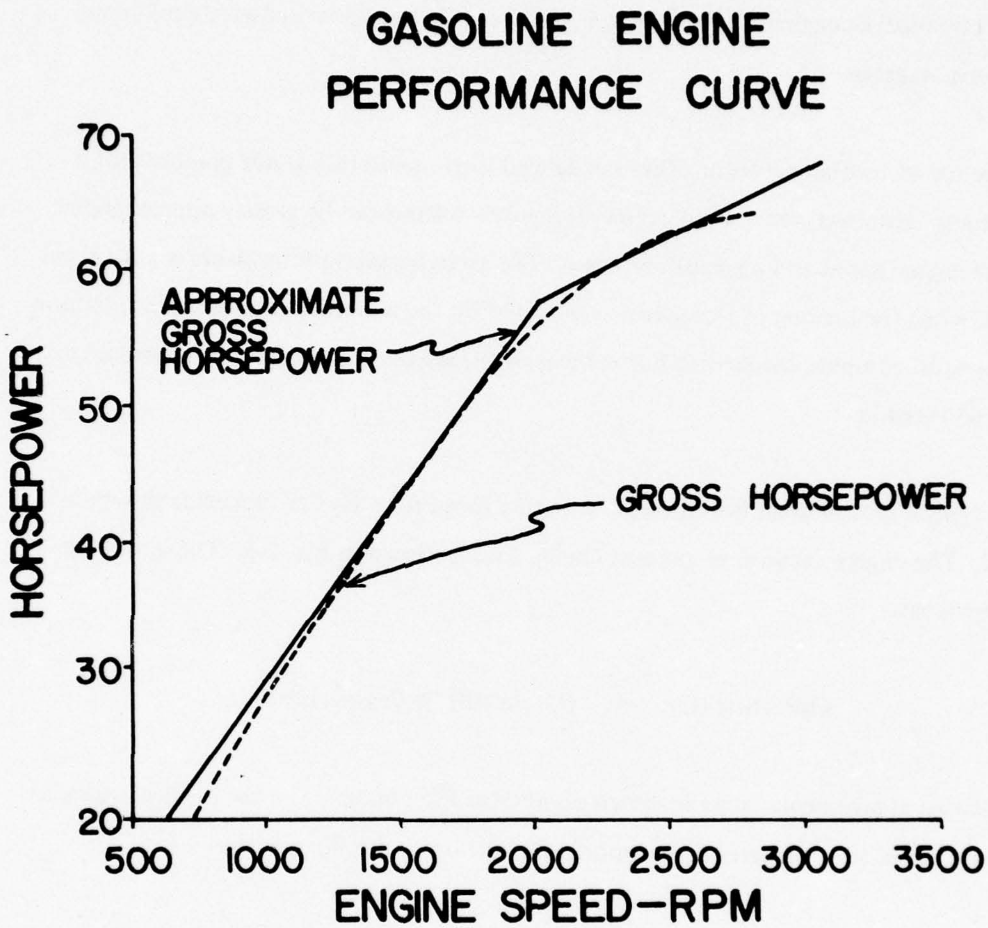

Fig. 2-2. Gasoline Engine Performance Curve.

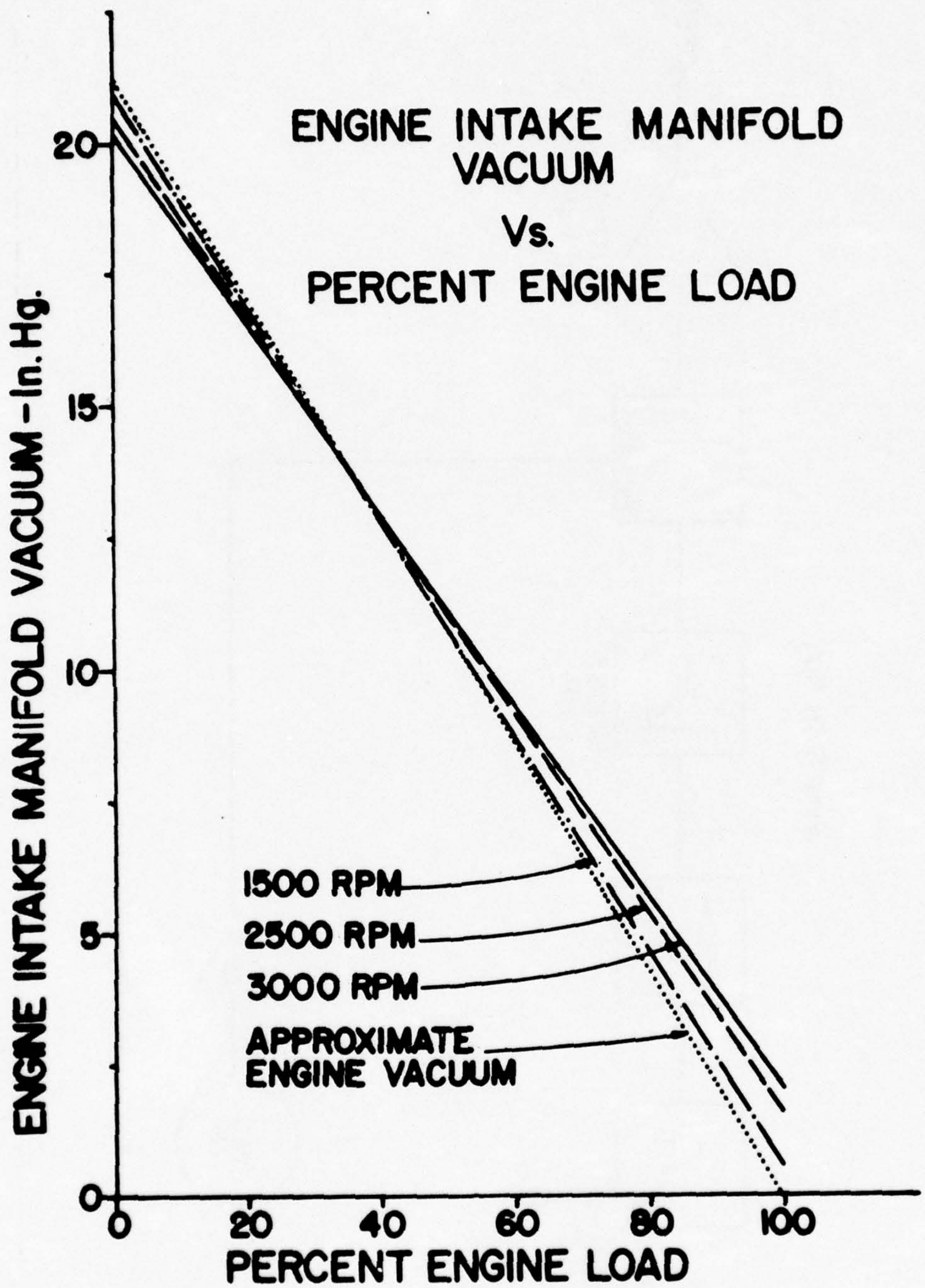

Fig. 2-3. Engine Intake Manifold Vacuum Vs. Percent Engine Load.

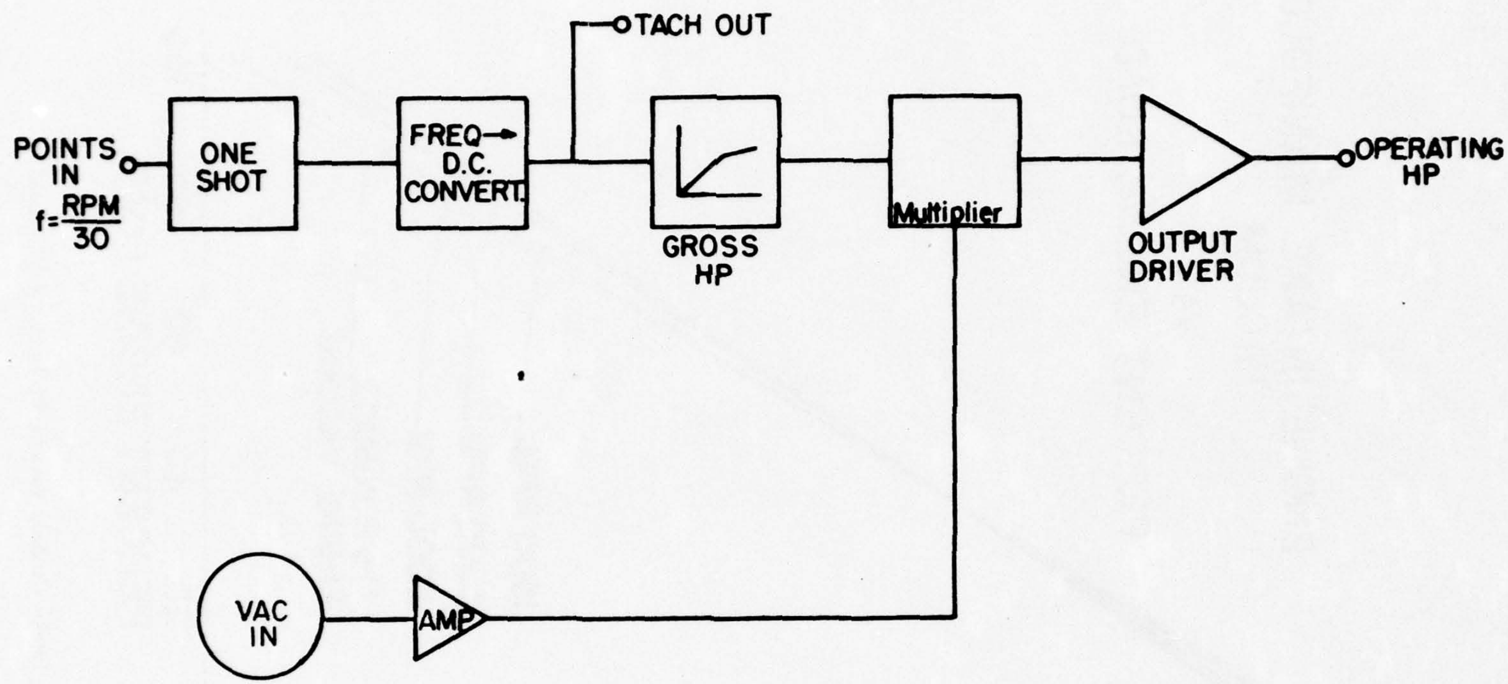

Fig. 2-4. Block Diagram.

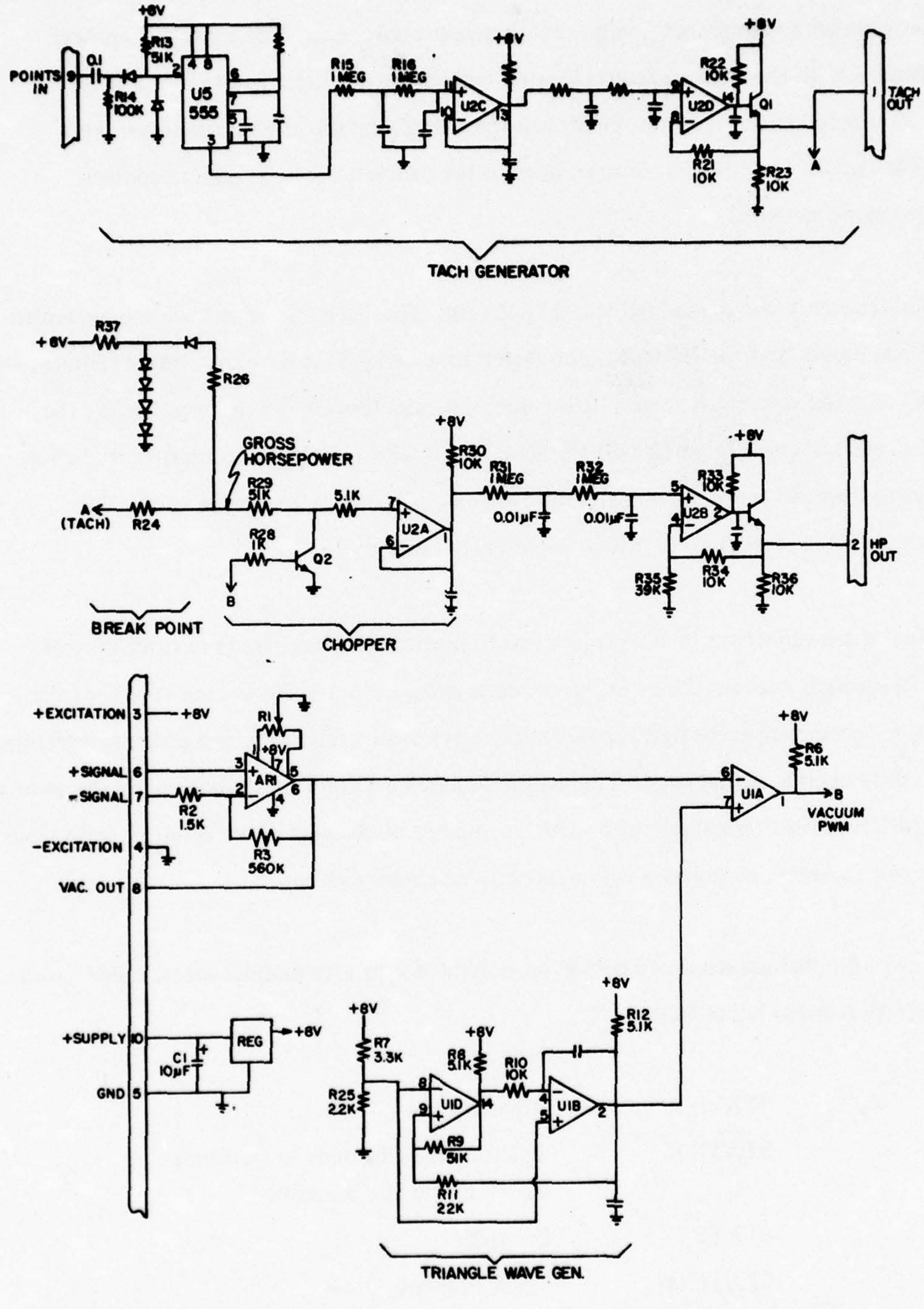

Fig. 2-5. Horsepower Computer Circuit.

converted to gross horsepower by the break point circuit. This circuit is a diode function generator which implements the required nonlinear conversion. The gross horsepower is multiplied with the engine vacuum by first converting the vacuum signal to a pulse width modulated signal (vacuum PWM) and then chopping the gross horsepower with this PWM signal. The filtered chopper signal is the product of the gross horsepower and the engine vacuum.

This circuitry was connected to a STAM unit. The Tach Out signal was connected to one STAM board, and the HP Out is connected to another STAM board. Initial field testing revealed that the concept is valid, but the design parameters are not appropriate for the intended applications. In particular, the signal band width of the HP computer is too low. In order to improve this band width, improved electronics will have to be used. This necessitates a new design cycle, which is presently underway.

One of the objectives of the project was to fabricate multi-channel readout devices (Type III), which were described in the previous annual report. During the course of the project, it became apparent that the increasing work load would require a printing interrogator which could operate unattended. This unit designated a Type IV, is functionally the same as the Type III with a thermal printer added. A number of changes in the circuitry were made to improve accuracy, including a crystal controlled timing system.

The multi-channel readout's operation is divided into four distinct areas, called *"states."* The operation is divided as follows:

STATE A	–	Resetting the E Cells
STATE B	–	Scanning the Channels to Determine Which E Cell Has Readout
STATE C	–	Printing
STATE D	–	Reset Finished

Transitions between the states are determined by the binary input signals listed below:

SIGNAL	*MEANING*
N_1	Some channel has readout 1 ⇒ true.
N_2	All channels have readout 1 ⇒ true.
N_3	Channel advance enables 1 ⇒ true.
N_4	Auto/man 0 ⇒ auto.
N_5	Particular channel addressed by 1 ⇒ true; the multiplexer is readout.
N_6	All channels have been checked 1 ⇒ true.

Fig. 2-6, known as a state diagram, describes the actual operation of the readout. The arrows represent transitions between the states, and the corresponding input tells when the transition occurs. The X's appearing in the inputs correspond to *"don't cares."*

This simply means the transition occurring, at that time, does not depend on that particular input signal; consequently, you don't care if the input is 0 or 1.

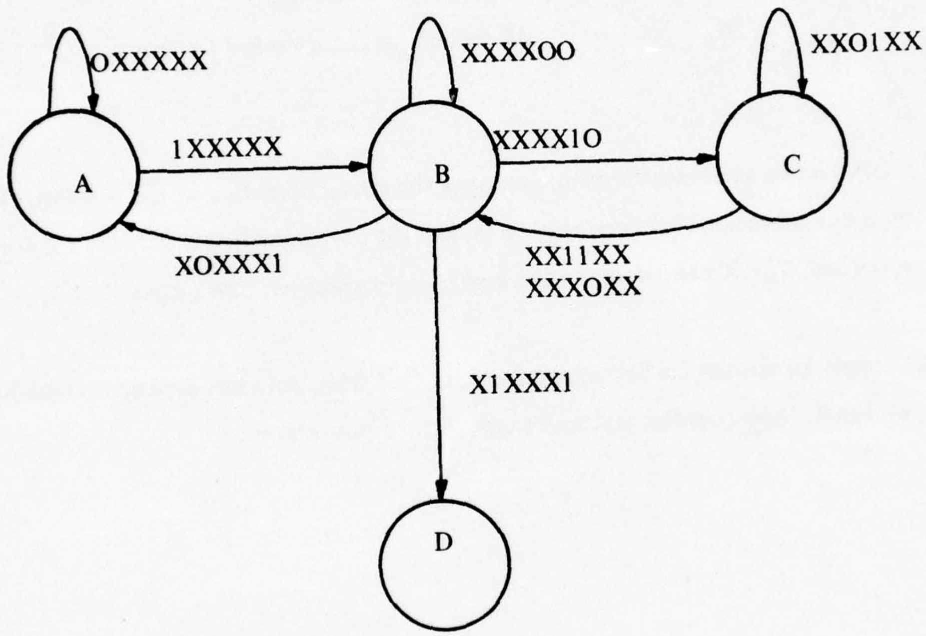

Fig. 2-6. Interrogator State Diagram.

As an example of how to read the state diagram, consider the initial state to be A and the inputs to be $N_1 = 0; N_2, N_3 \ldots N_6 = X$. The state diagram says remain in State A. If the input changes to $N_1 = 1; N_2, N_3 \ldots N_6 = X$ (A channel reads out.), a transition occurs to State B. While in State B, the channels are sequentially checked to determine which have readout. When the appropriate channel is found ($N_5 \rightarrow 1$), a transition to State C occurs. While in State C, printing will occur if the auto/manual switch is set to Auto ($N_4 = 0$). The operation is interrupted until the printing is completed. A transition then occurs back to State B to check the remaining channels. If the manual mode is selected ($N_4 = 1$), the operation is halted in State L to allow the operator to write down the information. A transition to State B will occur when the operator pushes the channel advance switch ($N_3 \rightarrow 1$). When all channels have been checked ($N_6 \rightarrow 1$), a transition occurs either to State A or State D. The decision between A and D is made by N_2. If all the channels have not readout ($N_2 = 0$), State A is chosen, and the readout operation continues. If all the channels have readout ($N_2 = 1$), State D is chosen and the readout operation terminated.

In State D, it is now possible to *"Set"* the timer E cell to the desired value. By use of logic simplification methods, the switching functions were determined, which allows the State Diagram to be implemented. Two of these multi-channel readouts are nearing completion at this time.

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CHAPTER III

DATA ACQUISITION

The availability of a broad base of operating data from fluid power systems would greatly enhance the capabilities of the industry to produce reliable systems. The earliest STAM field tests indicated that many of the widely held opinions regarding temperature and pressure duty cycles were incorrect. One of the principle aims of the data acquisition program was to determine the characteristic duty cycles of various machines (e.g., backhoes, loaders, dozers, etc.) and to identify the similarities associated with locale, system configuration, and type of work assignment. In order to achieve this goal efficiently, the monitors would be installed on a large number of machines of similar type. Specifically, all participants were asked to monitor either a loader or a backhoe. When the program was started, this was easily obtained, and the first 10 to 12 units were originally assigned to one of these types of vehicles. Only a few of these tests were completed. A widespread reduction in testing expenditures and manpower dictated a change in program constraints. In order to obtain any significant participation, the sponsors were permitted to install the units on any type vehicle which was used by the military.

The participating companies generally believe that they fully understand their existing systems. Therefore, there was a widespread desire to measure system parameters which were not previously explored. This further eroded the concept of a standard data base; but, in the end, it resulted in some very exciting new information. An installation in the suspension unit of a heavy duty truck proved extremely enlightening. Although the engineers involved were virtually certain that pressures never exceeded 7-8,000 psi, a 20,000 psi transducer was specified. The resulting data clearly revealed pressure pulses above 20,000 psi. In order to discount the possibility of electromagnetic interference, the STAM was installed with the pressure transducer not inserted in a hydraulic line. Some counts resulted from vibration and noise, but no time was recorded. Such noise very closely approximates the theoretical impulse (i.e., zero time). Thus, the time accumulated when the transducer was actually in the system makes it virtually certain that the hydraulic pressure did reach the higher levels.

In general, the data have shown more thermal and pressure cycles than were expected. The pressure cycles are subject to several potential errors. The most obvious is EMI, which can accumulate counts at the maximum frequency capability of the monitor. In addition, the transducers are not completely insensitive to vibration. Finally, there is always a flow/pressure ripple in the output of positive displacement pumps, in some cases of a very large magnitude.

The nature of the data can identify certain types of problems. The severe EMI conditions (such as the hour meter pulses) will activate all counters, resulting in a uniform high count at all levels. This noise is of such a short duration that the time accumulated at the higher gates is essentially zero. If the problem is caused by pump ripple or vibration, the spurious signal is superimposed on the basic transducer output. The counts recorded are caused by the signal rising and falling through a gate which nearly coincides with the mean signal value. It is almost impossible to distinguish such a situation from a rapidly cycling system (such as the load sensing system) without real-time data for reference.

The thermal cycles are not easily explained. The internal circuitry of the temperature boards is quite different than the pressure monitors and is much less susceptible to noise. There is no sensitivity to pressure ripple or vibration. Neither laboratory or field tests (including the hour meter tests) have been able to produce a spurious noise count. The only way known to produce false counts is to make and break the transducer connection which does occur when the transducer fails. Thus, it appears that, in general, there are far more thermal cycles in fluid power systems than had been expected. This will be a major focal point of future investigations.

As often happens with a new concept, there has been a tendency among some users to regard STAM as a replacement for all data recording methods. Every scheme for measuring involves trade-offs, and a realistic assessment of the most desirable approach to a given situation must consider opposing constraints.

The ultimate data acquisition system would be small enough to conceal in the system, self-powered, able to continuously record every variable for an infinite length of time, totally immune to EMI, and very low in cost. It would possess unlimited frequency response characteristics and interface easily with a computer for analysis of the data. There is no such system, and until STAM was developed the only course of action was to take short samples of real-time data collected with expensive and fairly bulky equipment.

STAM was conceived not to replace this equipment but to help fill the gap between present capabilities and the ultimate system. STAM allows long-term, low-cost, statistical evaluations of the system without limiting or intruding upon the system capability.

Nearly all of the limitations of STAM can be overcome in a properly conducted total data acquisition system. The scaling factors, gate levels, and hysteresis value can be modified on the monitor if a short sample of real-time data is used to establish the proper adjustments.

An interesting evaluation of the potential of Statistical Monitoring is shown in Fig. 3-1. An extensive real-time recording of system pressure was obtained by Clark Equipment Company. This tape was processed by a computer to simulate a STAM recording. The STAM data were then reconstructed into an equivalent wave form. The wave form shown is the block form (no smoothing applied), but it offers a valid comparison of the two recording methods.

The most instructive data sets obtained from the industrial equipment monitoring program are presented in Appendix A. The information from MERDC operated units is not included, as these monitors are interrogated at the test location.

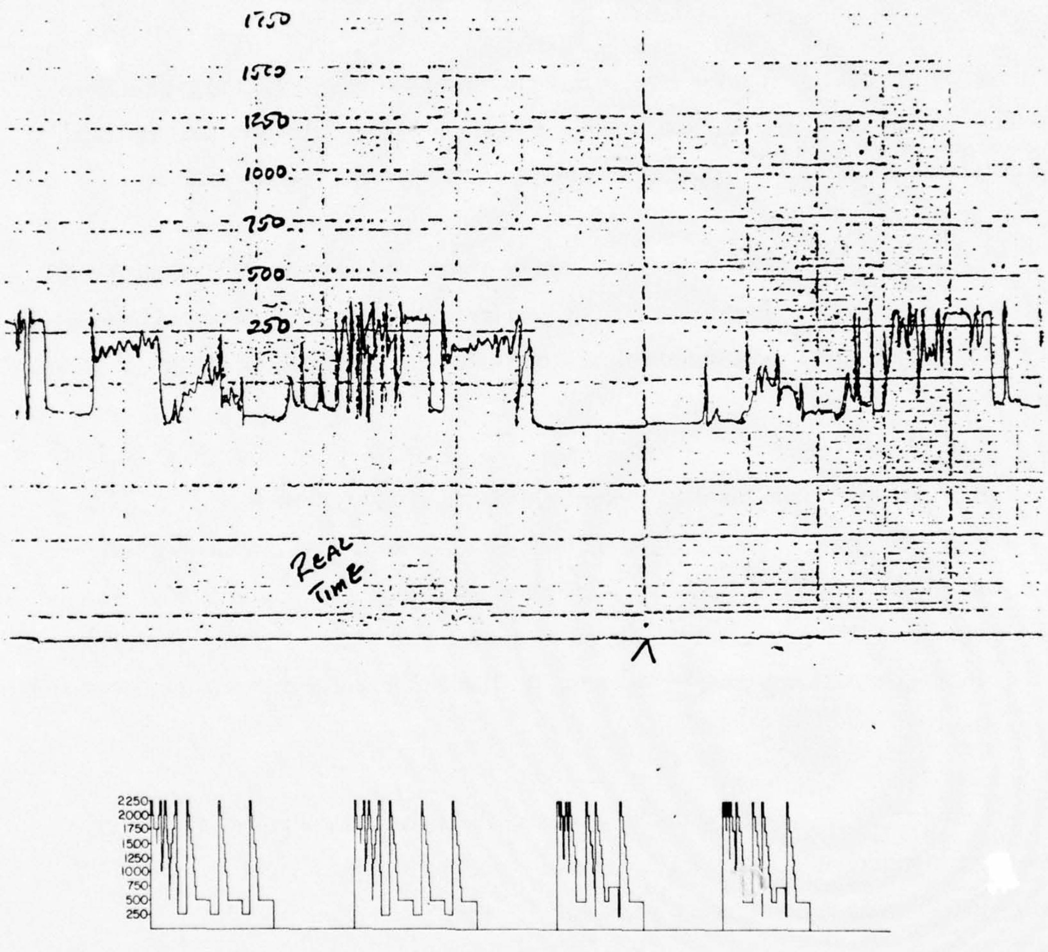

Fig. 3-1. Comparison of Real-Time and Reconstructed Duty Cycles.

CHAPTER IV

SUMMARY & CONCLUSIONS

The use of Statistical Monitoring to obtain an understanding of the operational severity of fluid power systems has been shown to be a viable concept. The reliability of the hardware and the usefulness of the data have been accepted by the industrial sponsors to such an extent that the demand for monitors exceeds the availability of manpower to support the effort. The supreme test in the industrial environment is the willingness of the users to support the program financially. This test has been met. The direct industrial sponsorship for the next year is already in excess of the funds scheduled on this contract, and additional sponsor participation is virtually certain. This is a vital consideration in achieving the goal of a broad information base. The participation of sponsor personnel and test equipment in the past have been very costly to the sponsors and very valuable to the program. However, it must be recognized that the bulk of the quality information came where there was a high degree of visibility of the program within the company. Such visibility is assured when there is a direct expenditure of funds.

In the early stages of the monitor program, it was believed that once the basic Statistical Monitors had been completed that the hardware development would cease. This is unrealistic in view of the dynamic state of technology and the increasing number of applications for STAM. The project group has been reorganized to recognize this fact; and, in the future, the hardware activities should compliment rather than interfere with the analytical aspects.

The rapidly awaking interest in Statistical Monitoring and the availability of adequate funding, together with the experienced research, engineering, and technical staff should promote rapid progress in the developing of a fuller understanding of fluid power system duty cycles.

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APPENDIX A

DATA SHEETS

DATA SHEET

DATE: 22 January 1976 **UNIT NO.:** 1030–130

COMPANY: 1

UNIT TYPE: Pressure/Temperature 130 = Pressure

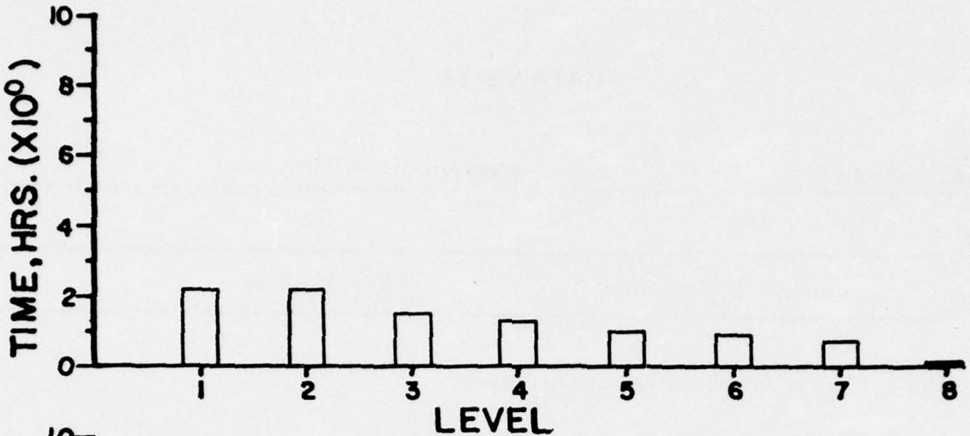
APPLICATION: *(Type of Vehicle and Location of Sensors)*

Excavator, Inlet to Dig/Hoist Valve, Northeast U.S.

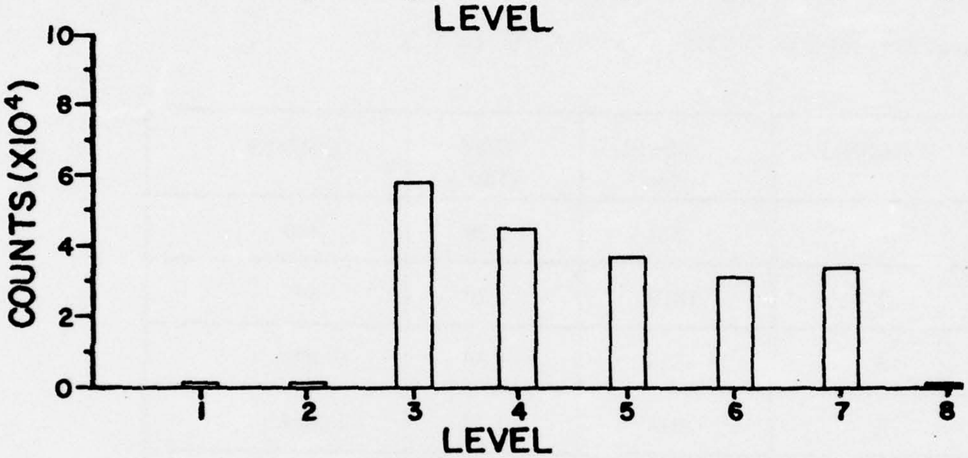
CHANNEL	LEVEL (psi)	TIME (HRS.)	COUNTS
1	521	2.20	460
2	1019	2.20	445
3	1528	1.50	57,890
4	2038	1.24	45,014
5	2547	1.03	36,373
6	3057	.88	31,194
7	3555	.74	34,088
8	4064	.12	349

5.6 **(Hrs.) Total Operation Time**

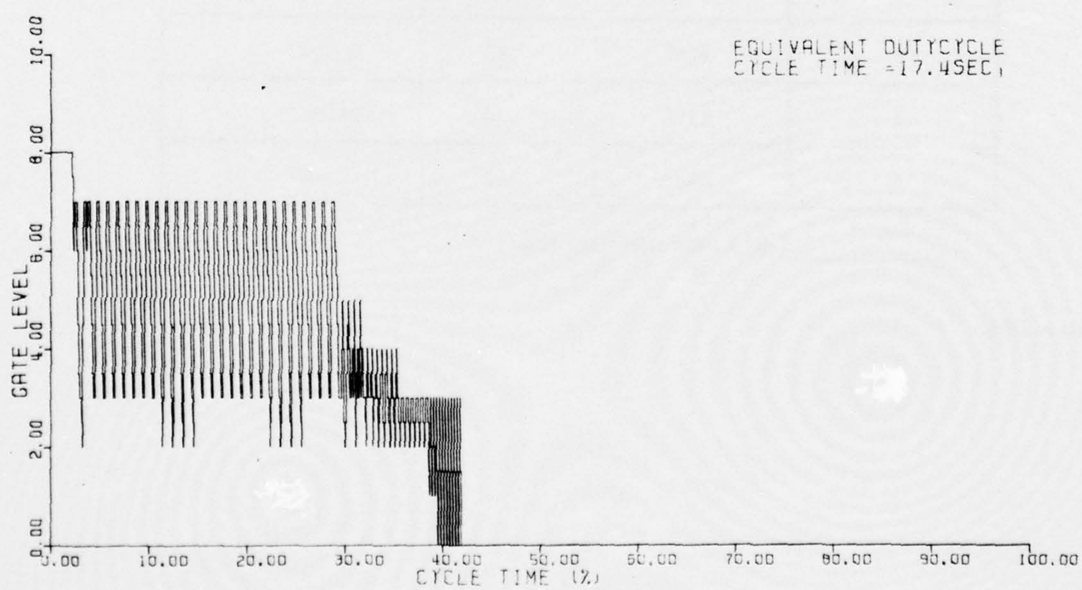
REMARKS:



TIME, HRS. (X10⁰)
10
8
6
4
2
0
1
2
3
4
5
6
7
8
LEVEL



COUNTS (X10⁴)
10
8
6
4
2
0
1
2
3
4
5
6
7
8
LEVEL



EQUIVALENT DUTYCYCLE
CYCLE TIME =17.45EC
GATE LEVEL
0.00
2.00
4.00
6.00
8.00
10.00
0.00
10.00
20.00
30.00
40.00
50.00
60.00
70.00
80.00
90.00
100.00
CYCLE TIME (%)

DATA SHEET

DATE: 22 January 1976 **UNIT NO.:** 1030–129

COMPANY: 1

UNIT TYPE: Pressure/Temperature Temperature = 129

APPLICATION: *(Type of Vehicle and Location of Sensors)*

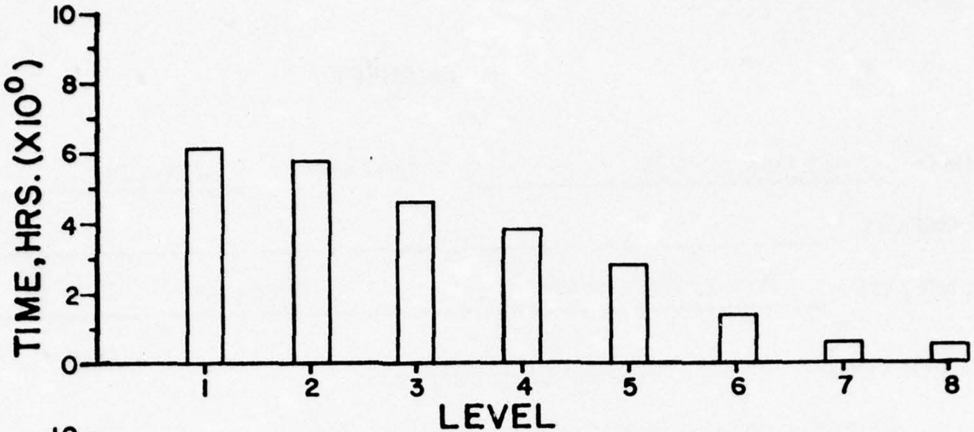
Excavator, Inlet to Dig/Hoist Valve, Northeast U.S.

CHANNEL	LEVEL (°F)	TIME (HRS.)	COUNTS
1	78.0	6.07	11,486
2	98.8	5.76	18,247
3	120.0	4.59	22,112
4	140.8	3.81	14,227
5	161.6	2.79	10,313
6	181.8	1.40	7,933
7	203.0	.54	3,456
8	223.8	.57	2,241

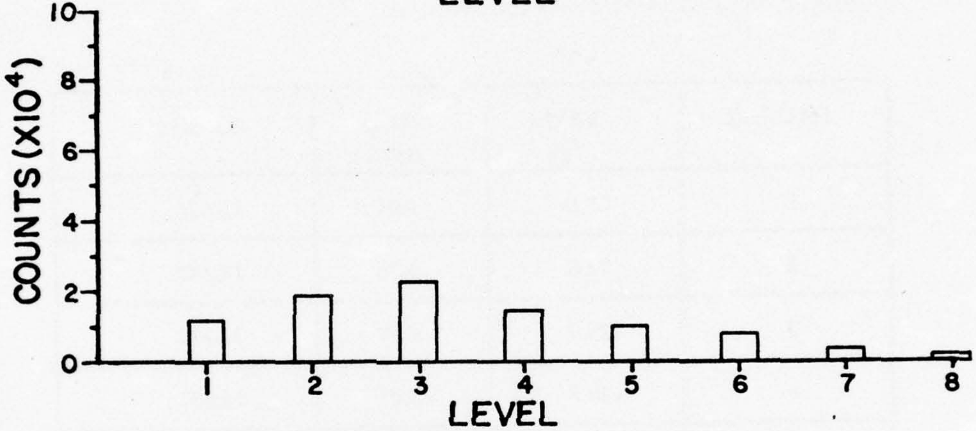
6.07 **(Hrs.) Total Operation Time**

REMARKS:

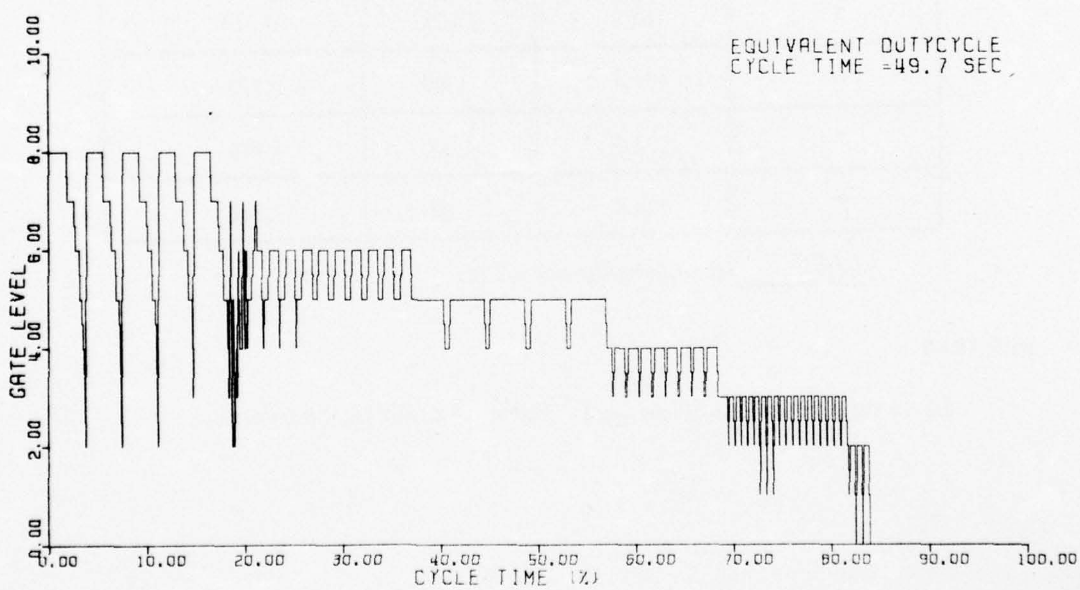
Large number of counts suspect. Noise filter added for next test.



TIME, HRS. (X10⁰)
10
8
6
4
2
0
1
2
3
4
5
6
7
8
LEVEL



COUNTS (X10⁴)
10
8
6
4
2
0
1
2
3
4
5
6
7
8
LEVEL



EQUIVALENT DUTYCYCLE
CYCLE TIME =49.7 SEC
GATE LEVEL
10.00
8.00
6.00
4.00
2.00
0.00
0.00
10.00
20.00
30.00
40.00
50.00
60.00
70.00
80.00
90.00
100.00
CYCLE TIME (%)

DATA SHEET

DATE: 1 October 1974 **UNIT NO.:** 1015

COMPANY: 2

UNIT TYPE: Temperature

APPLICATION: *(Type of Vehicle and Location of Sensors)*

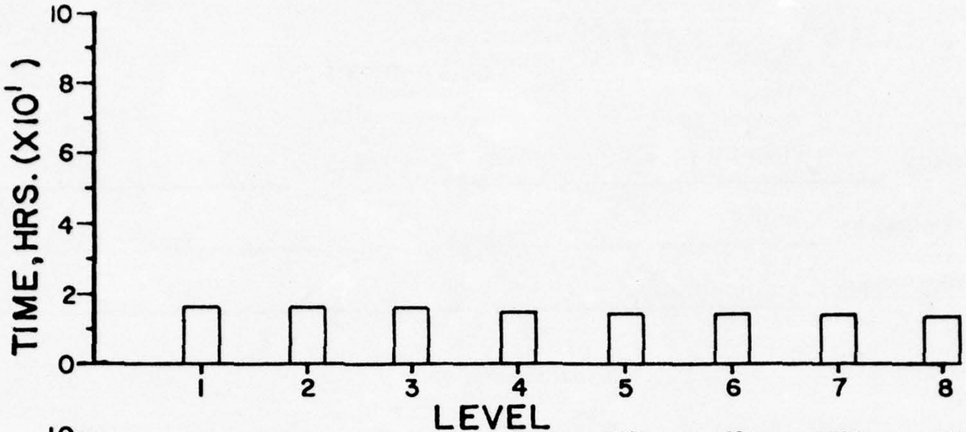
Tractor (Wheel Type), Main System Flow, North Central U.S.

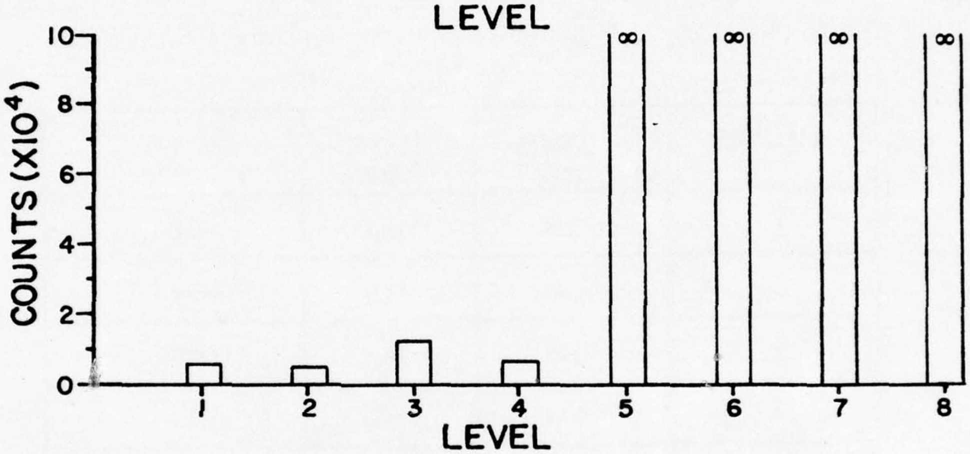
CHANNEL	LEVEL (°F)	TIME (HRS.)	COUNTS
1	78	16.2	5081
2	98	15.9	4749
3	119	15.9	11,877
4	139	14.6	5715
5	159	14.2	∞
6	179	14.1	∞
7	199	14.1	∞
8	220	13.8	∞

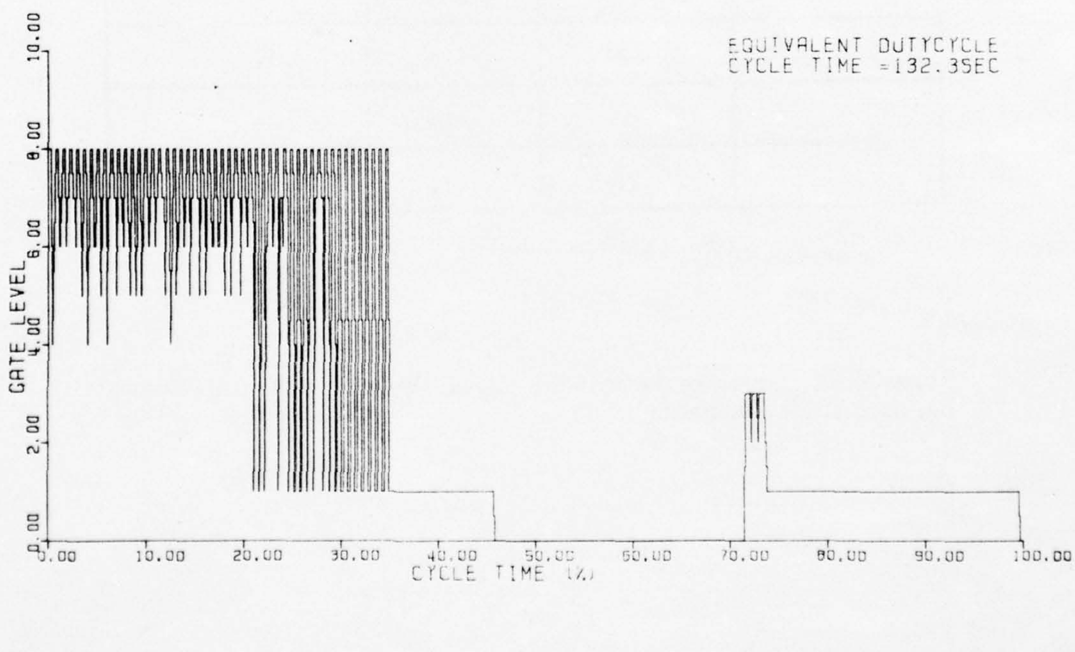
73.5 **(Hrs.) Total Operation Time**

REMARKS:

Temperature transducer destroyed, replaced. Sponsor verified transducer damaged at 20.2 operating hours.


TIME, HRS. (X10¹)
LEVEL


COUNTS (X10⁴)
∞
LEVEL


EQUIVALENT DUTYCYCLE
CYCLE TIME =132.3SEC
GATE LEVEL
CYCLE TIME (%)

DATA SHEET

DATE: 25 April 1975 UNIT NO.: 1015

COMPANY: 2

UNIT TYPE: Temperature

APPLICATION: *(Type of Vehicle and Location of Sensors)*

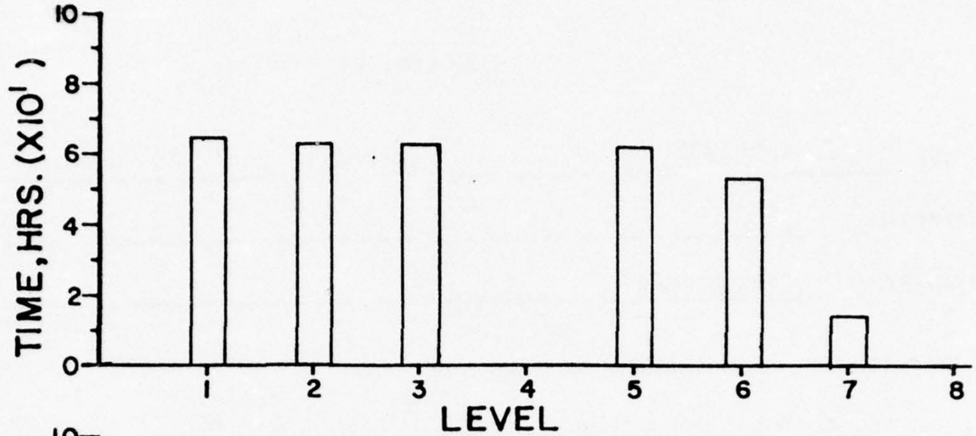
Tractor (Wheel Type), Main System Flow, North Central U.S.

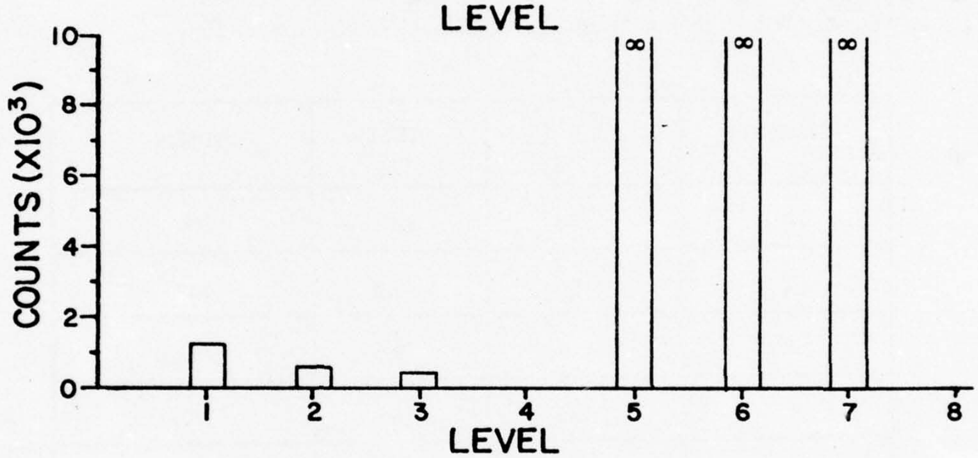
CHANNEL	LEVEL (°F)	TIME (HRS.)	COUNTS	
1	78	64.3	1218	
2	98	63.3	547	
3	119	63.0	453	
4	139	0	---	--- Defective Timer
5	159	61.7	∞	
6	179	53.2	∞	
7	199	14.1	∞	
8	220	0	0	

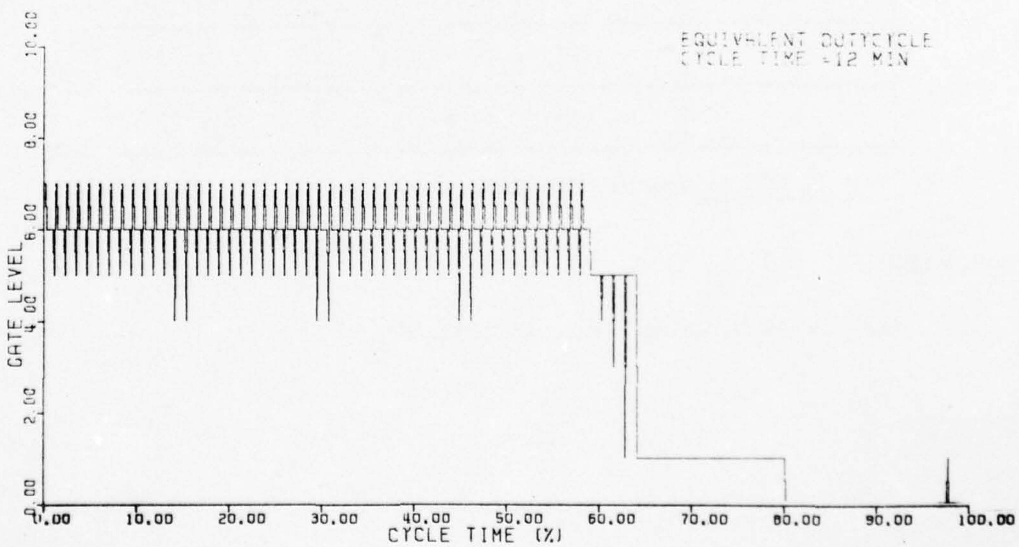
100 (Hrs.) Total Operation Time

REMARKS:

Temperature transducer destroyed, replaced.


TIME, HRS. (X10¹)
10
8
6
4
2
0
1 2 3 4 5 6 7 8
LEVEL


COUNTS (X10³)
10
8
6
4
2
0
∞ ∞ ∞
1 2 3 4 5 6 7 8
LEVEL


EQUIVALENT DUTYCYCLE
CYCLE TIME =12 MIN
GATE LEVEL
10.00
8.00
6.00
4.00
2.00
0.00
0.00 10.00 20.00 30.00 40.00 50.00 60.00 70.00 80.00 90.00 100.00
CYCLE TIME (%)

DATA SHEET

DATE: 7 July 1975 **UNIT NO.:** 1015

COMPANY: 2

UNIT TYPE: Temperature

APPLICATION: *(Type of Vehicle and Location of Sensors)*

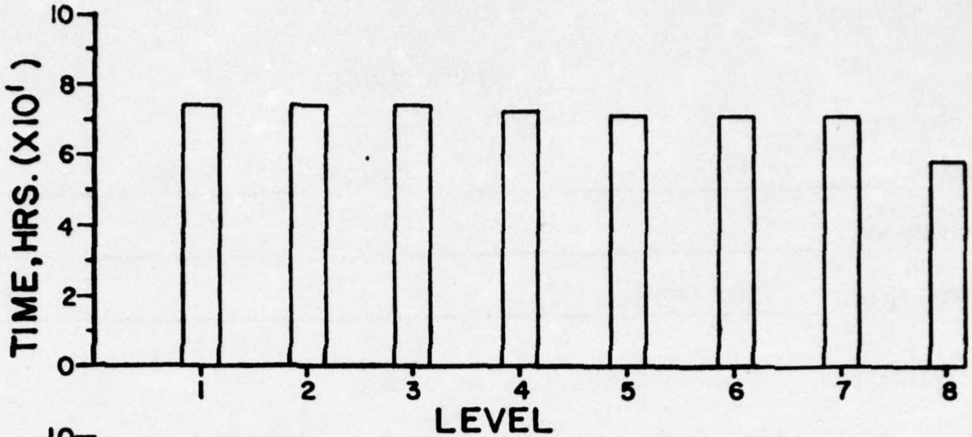
Tractor (Wheel Type), Main System Flow, North Central U.S.

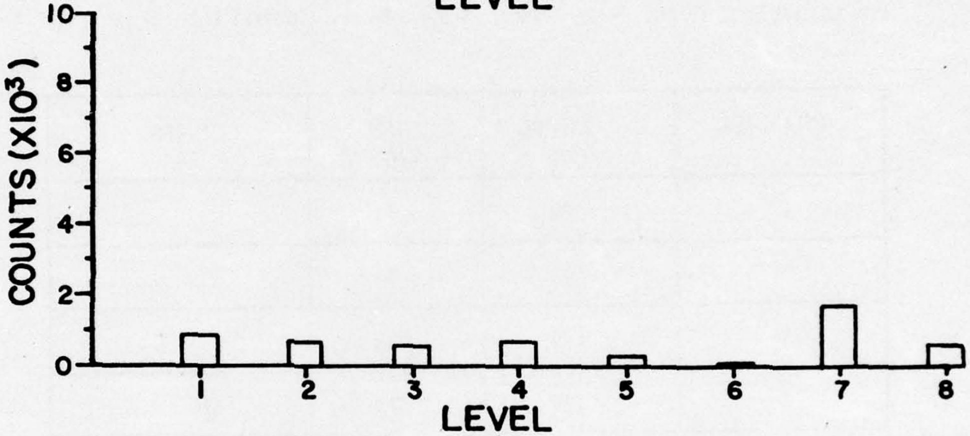
CHANNEL	LEVEL (°F)	TIME (HRS.)	COUNTS
1	78	74	859
2	98	74	625
3	119	74	533
4	139	72	619
5	159	71	210
6	179	71	54
7	199	71	1664
8	220	58	478

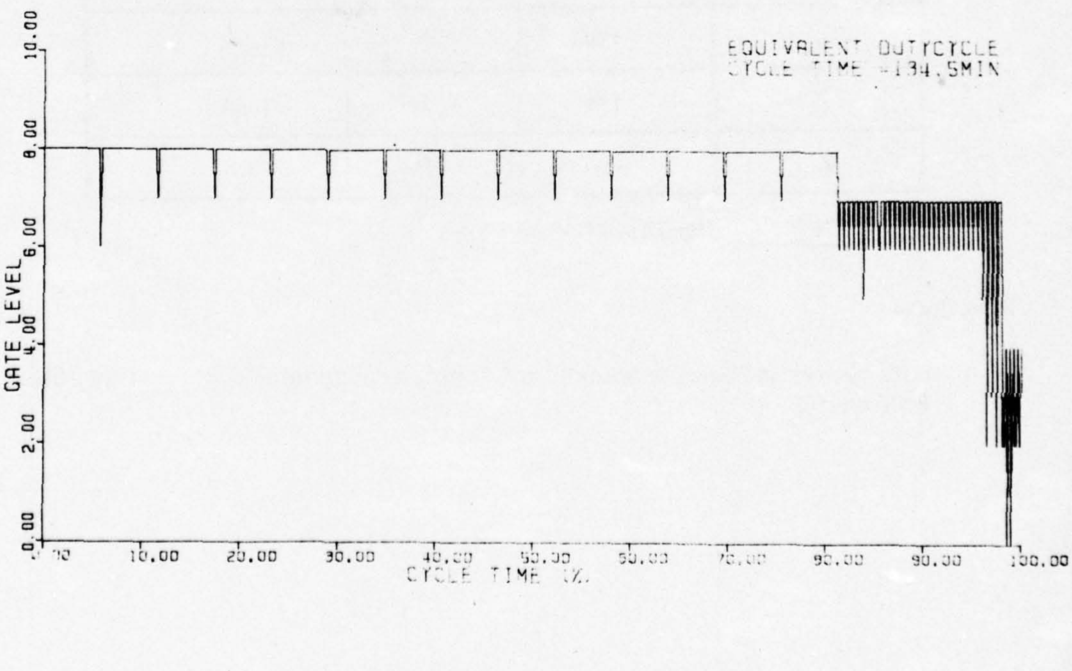
74 **(Hrs.) Total Operation Time**

REMARKS:

Data pattern indicates transducer not connected. Sponsor confirmed 24 July. Unit retired.


TIME, HRS. (X10¹)
LEVEL
0
2
4
6
8
10
1
2
3
4
5
6
7
8


COUNTS (X10³)
LEVEL
0
2
4
6
8
10
1
2
3
4
5
6
7
8


EQUIVALENT DUTYCYCLE
CYCLE TIME =134.5MIN
GATE LEVEL
CYCLE TIME (%)
0.00
2.00
4.00
6.00
8.00
10.00
10.00
20.00
30.00
40.00
50.00
60.00
70.00
80.00
90.00
100.00

DATA SHEET

DATE: 17 March 1975 **UNIT NO.:** 1020

COMPANY: 3

UNIT TYPE: Pressure

APPLICATION: *(Type of Vehicle and Location of Sensors)*

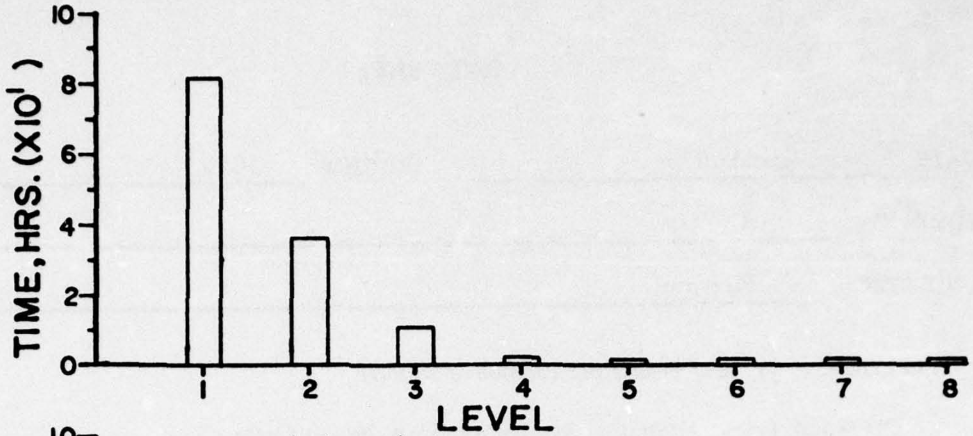
Off Road Truck, Hydraulic Suspension Unit, North Central U.S.

CHANNEL	LEVEL PSI	TIME (HRS.)	COUNTS
1	2500	81.0	1175
2	5000	36.0	> 75,000
3	7500	10.5	> 75,000
4	10,000	1.6	35,500
5	12,500	1.2	21,400
6	15,000	1.2	18,150
7	17,500	1.14	15,900
8	20,000	1.27	14,300

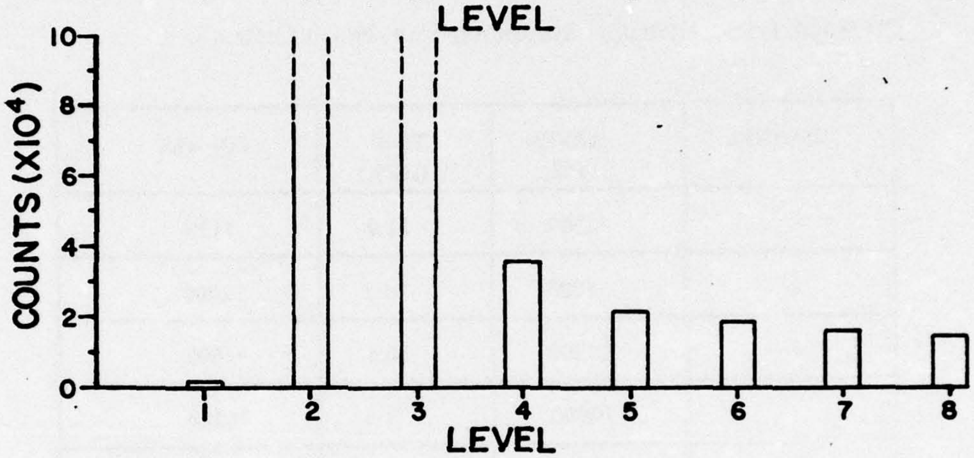
81 **(Hrs.) Total Operation Time**

REMARKS:

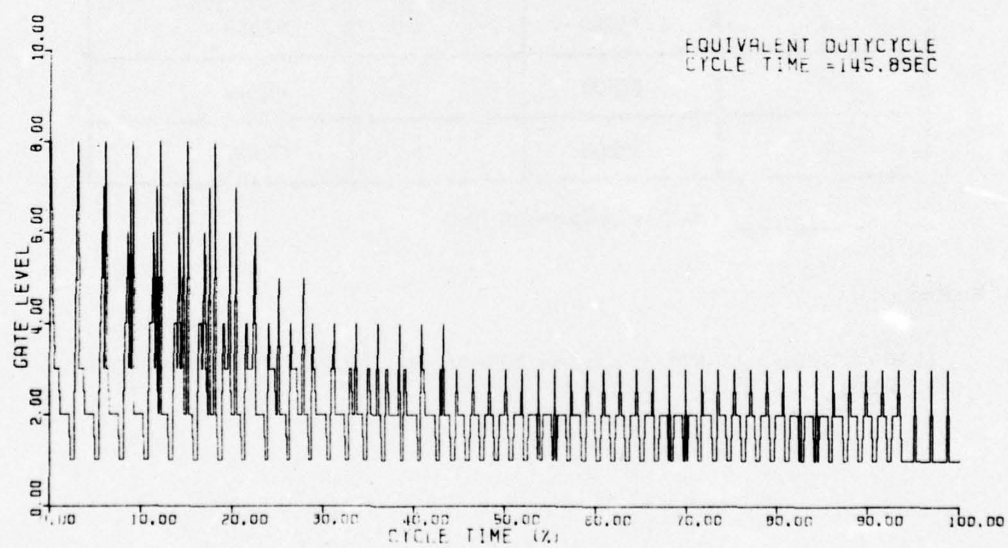
Data does not conform to sponsor analysis, but 10,000 psi rated component is failing.



TIME, HRS. (X10¹)
LEVEL



COUNTS (X10⁴)
LEVEL



EQUIVALENT DUTYCYCLE
CYCLE TIME =145.8SEC
GATE LEVEL
CYCLE TIME (%)

DATA SHEET

DATE: 20 May 1975 **UNIT NO.:** 1020

COMPANY: 3

UNIT TYPE: Pressure

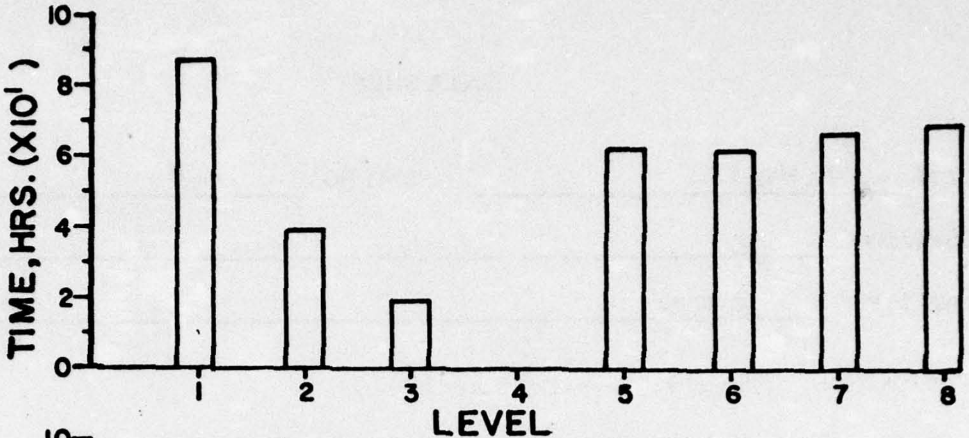
APPLICATION: *(Type of Vehicle and Location of Sensors)*

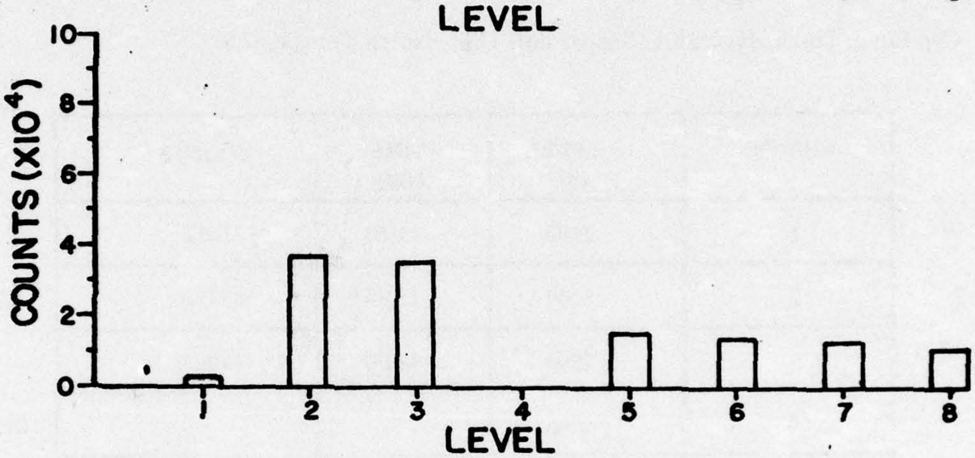
Off Road Truck, Hydraulic Suspension Unit, North Central U.S.

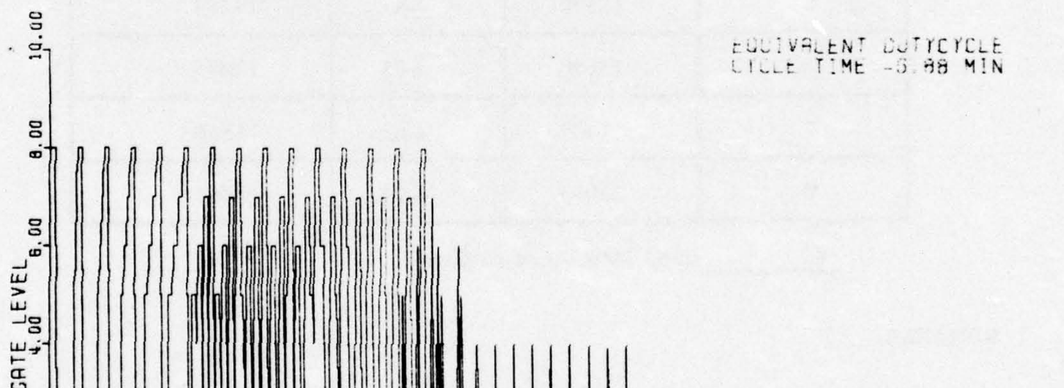
CHANNEL	LEVEL PSI	TIME (HRS.)	COUNTS	
1	2500	81.60	1652	
2	5000	35.90	35,766	
3	7500	11.90	34,045	
4	10,000	-----	-----	Defect
5	12,500	6.05	13,250	
6	15,000	6.05	12,895	
7	17,500	6.67	11,665	
8	20,000	6.74	10,482	

82 **(Hrs.) Total Operation Time**

REMARKS:


TIME, HRS. (X10^1)
LEVEL


COUNTS (X10^4)
LEVEL


EQUIVALENT DUTYCYCLE
GATE LEVEL
CYCLE TIME (%)

DATA SHEET

DATE: 5 January 1975 **UNIT NO.:** 1078–177

COMPANY: 2

UNIT TYPE: Pressure/Temperature 177 = Temperature

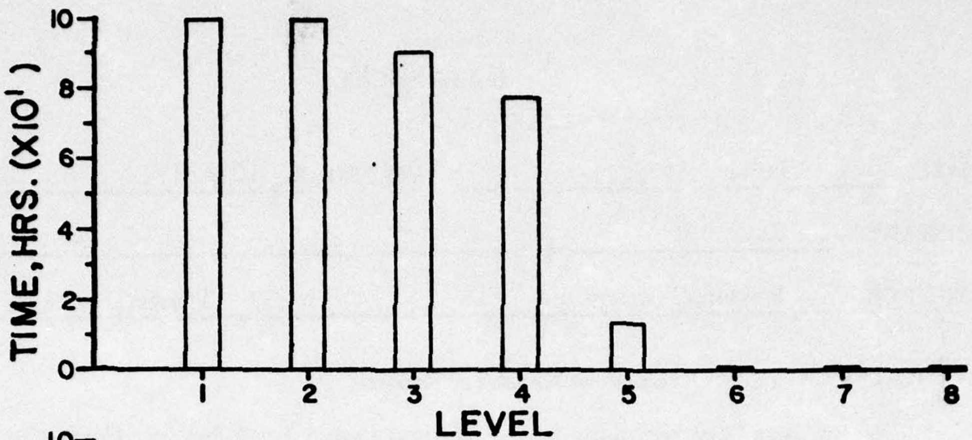
APPLICATION: *(Type of Vehicle and Location of Sensors)*

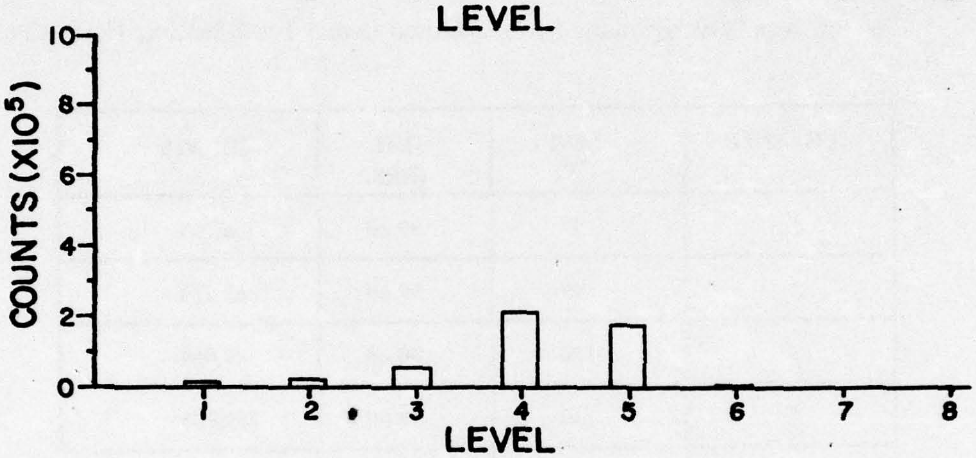
Tractor, Auxiliary Hydraulic System, Closed Center Load Sensing, North Central U.S.

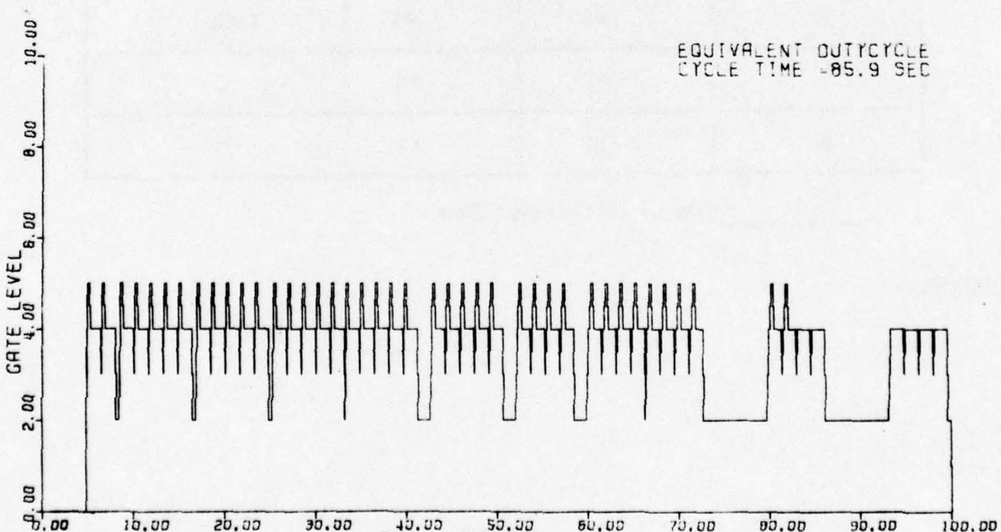
CHANNEL	LEVEL (°F)	TIME (HRS.)	COUNTS
1	78	99.80	6,208
2	99	99.60	13,274
3	120	90.40	45,064
4	140	77.80	> 209,000
5	161	14.10	183,193
6	181	.93	1,901
7	202	.84	0
8	223	.63	0

99.8 **(Hrs.) Total Operation Time**

REMARKS:


TIME, HRS. (X10¹)
LEVEL


COUNTS (X10⁵)
LEVEL


EQUIVALENT DUTYCYCLE
CYCLE TIME =85.9 SEC
GATE LEVEL
CYCLE TIME (%)

DATA SHEET

DATE: 5 January 1975 **UNIT NO.:** 1078–178

COMPANY: 3

UNIT TYPE: Pressure/Temperature Pressure = 178

APPLICATION: *(Type of Vehicle and Location of Sensors)*

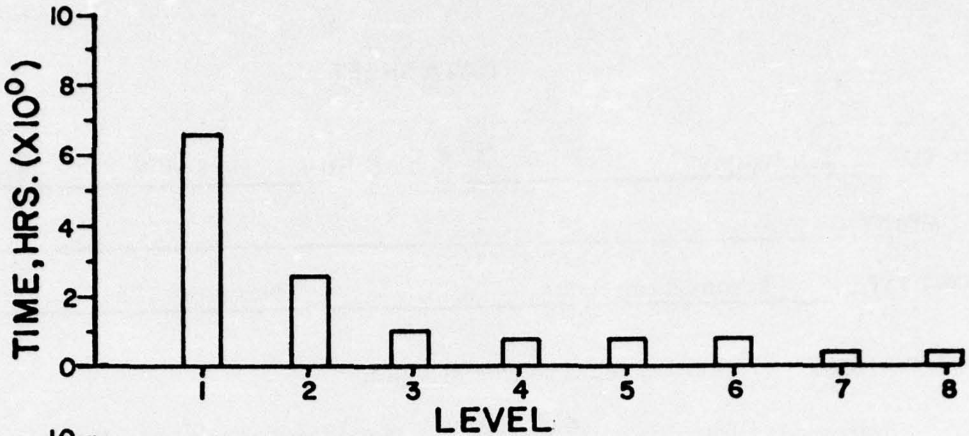
Tractor, Auxiliary Hydraulic System, Closed Center, Load Sensing, North Central U.S.

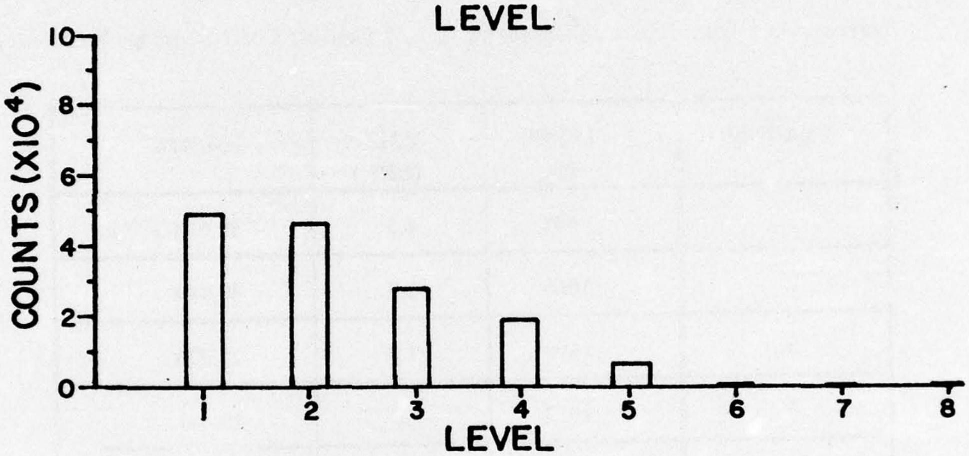
CHANNEL	LEVEL PSI	TIME (HRS.)	COUNTS
1	641	6.5	49,618
2	1065	2.5	49,340
3	1500	1.0	28,326
4	1923	.7	19,451
5	2348	.7	6,630
6	2772	.7	0
7	3196	.2	0
8	3630	.2	0

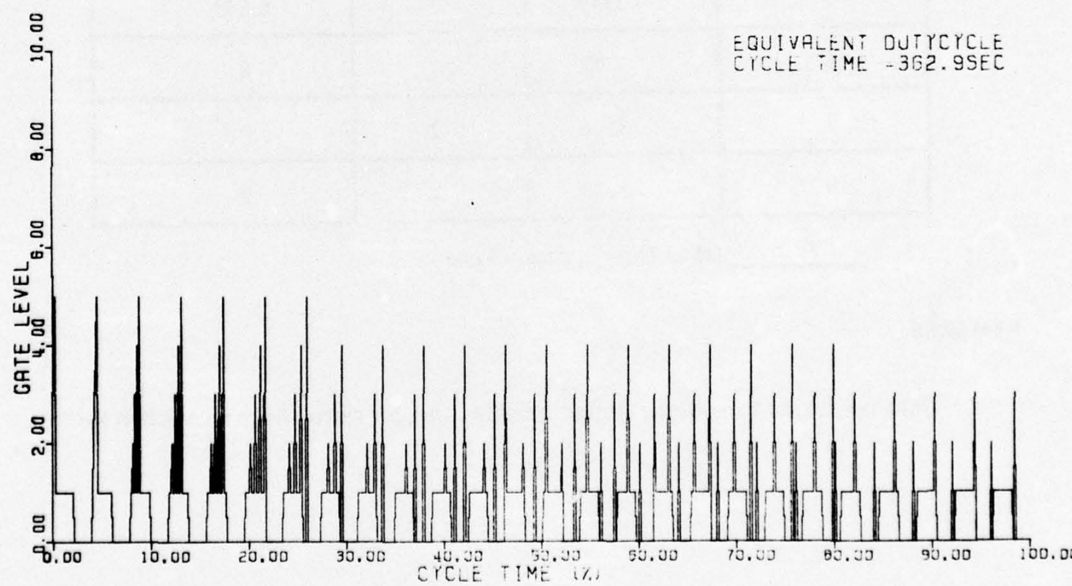
99.8 **(Hrs.) Total Operation Time**

REMARKS:

Data conforms to sponsor design analysis and previous short-term observations.


TIME, HRS. (X10⁰)
10
8
6
4
2
0
1
2
3
4
5
6
7
8
LEVEL


COUNTS (X10⁴)
10
8
6
4
2
0
1
2
3
4
5
6
7
8
LEVEL


EQUIVALENT DUTYCYCLE
CYCLE TIME -362.9SEC
GATE LEVEL
10.00
8.00
6.00
4.00
2.00
0.00
0.00
10.00
20.00
30.00
40.00
50.00
60.00
70.00
80.00
90.00
100.00
CYCLE TIME (%)

DATA SHEET

DATE: 19 December 1975 **UNIT NO.:** 1028–128

COMPANY: 4

UNIT TYPE: Pressure/Temperature Pressure = 128

APPLICATION: *(Type of Vehicle and Location of Sensors)*

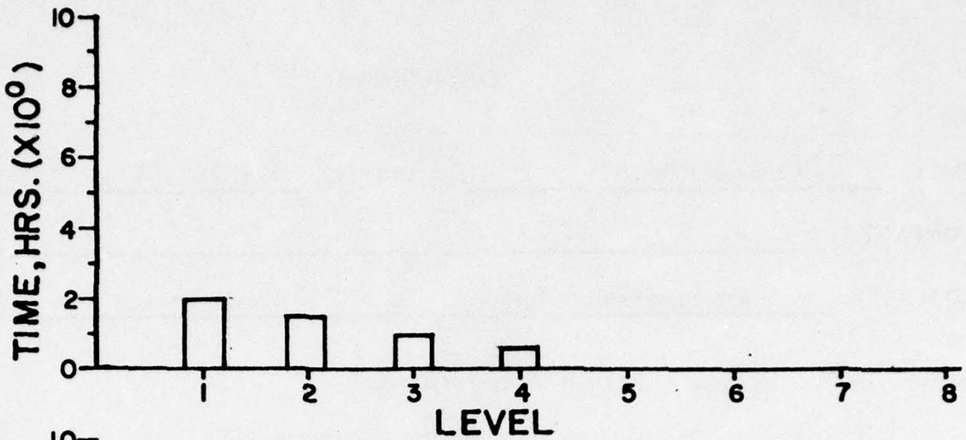
Industrial Backhoe Loader – Main System, Southwest U.S.

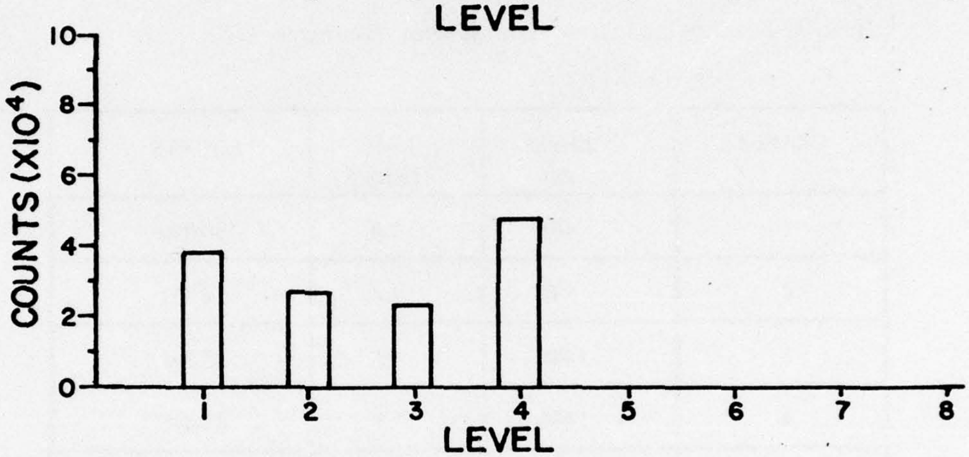
CHANNEL	LEVEL PSI	TIME (HRS.)	COUNTS
1	488	2.0	38,050
2	948	1.4	26,361
3	1408	.9	23,332
4	1858	.5	48,445
5	2318	0.0	
6	2778	0.0	
7	3238	0.0	
8	3688	0.0	

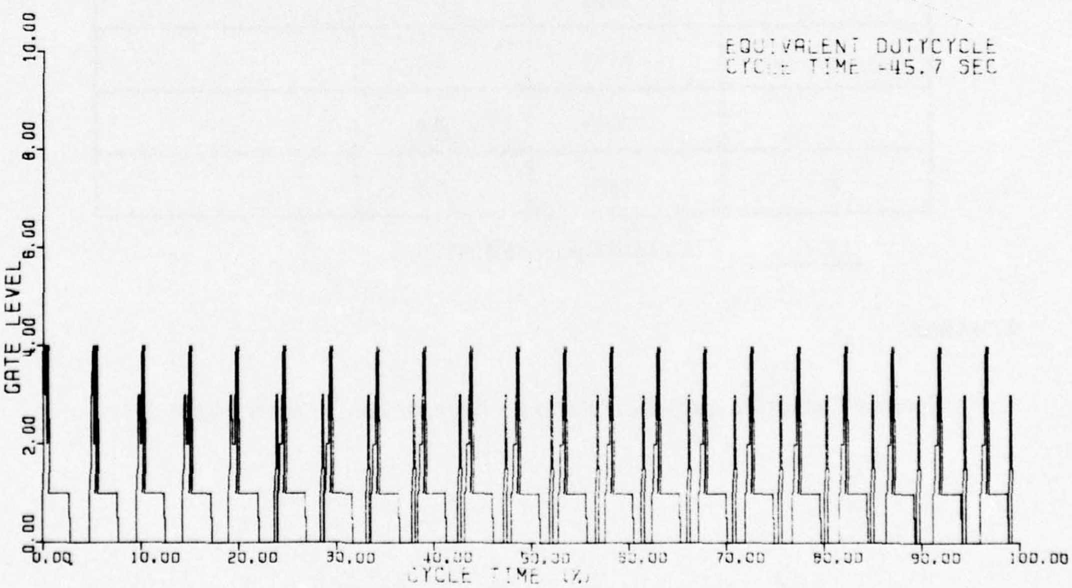
12.7 **(Hrs.) Total Operation Time**

REMARKS:

Sponsor furnished transducer leaking. Sponsor confirms very light duty.


TIME, HRS. (X10^{0})
LEVEL


COUNTS (X10^{4})
LEVEL


EQUIVALENT DUTYCYCLE
CYCLE TIME -45.7 SEC
GATE LEVEL
CYCLE TIME (%)

DATA SHEET

DATE: 19 December 1975 UNIT NO.: 1028–127

COMPANY: 4

UNIT TYPE: Pressure/Temperature Temperature = 127

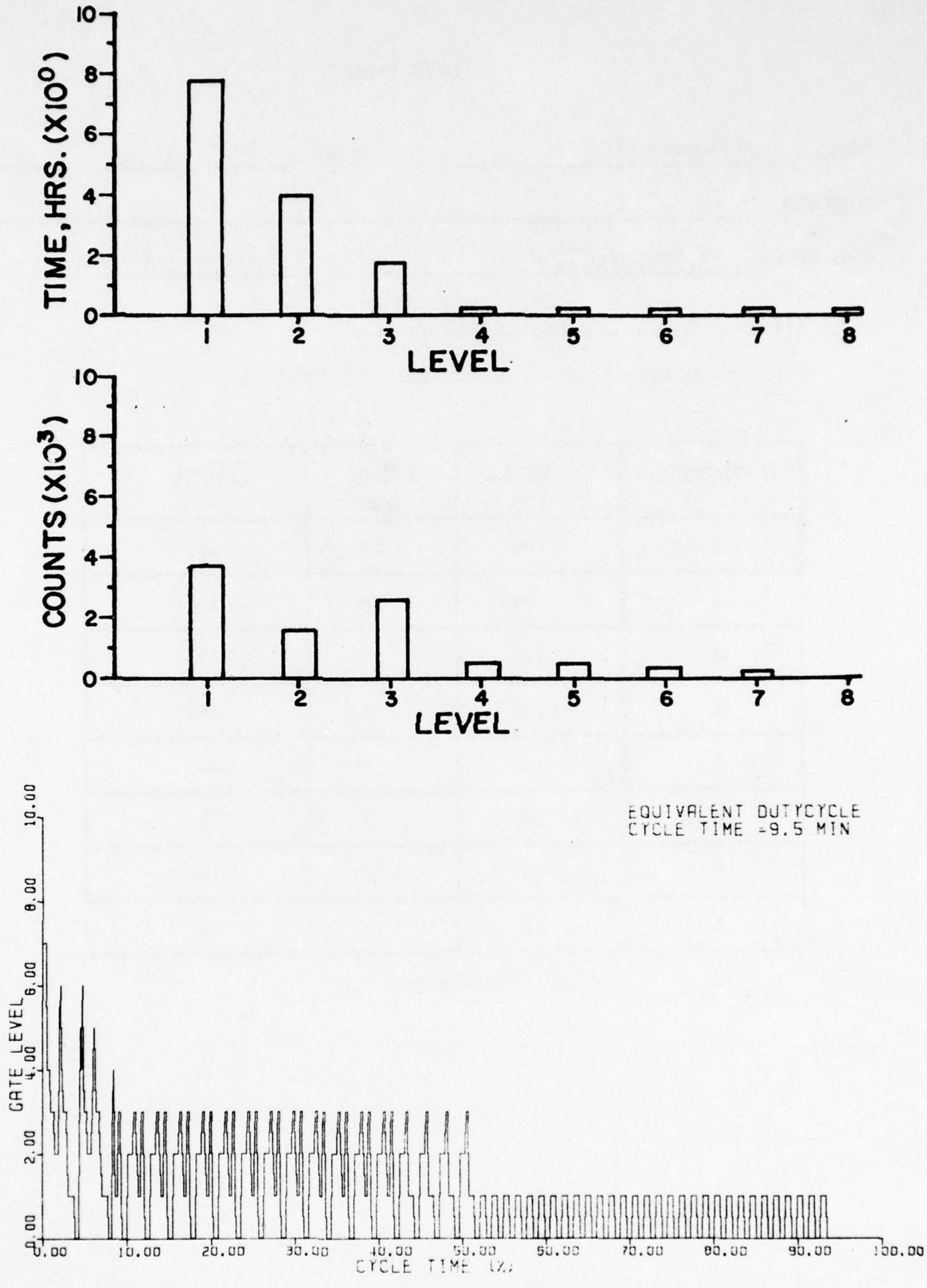
APPLICATION: *(Type of Vehicle and Location of Sensors)*

Industrial Backhoe Loader – Main System, Southwest U.S.

CHANNEL	LEVEL (°F)	TIME (HRS.)	COUNTS
1	78	7.8	3,892
2	99	4.0	1,536
3	120	1.6	2,693
4	140	.1	396
5	161	.1	318
6	181	.1	236
7	201	.1	104
8	222	.1	0

12.7 (Hrs.) Total Operation Time

REMARKS:



TIME, HRS. (X10⁰)
LEVEL
COUNTS (X10³)
LEVEL
EQUIVALENT DUTYCYCLE
CYCLE TIME =9.5 MIN
GATE LEVEL
CYCLE TIME (%)

DATA SHEET

DATE: 20 November 1975 UNIT NO.: 1042–141

COMPANY: 5

UNIT TYPE: Pressure/Temperature Temperature = 141

APPLICATION: *(Type of Vehicle and Location of Sensors)*

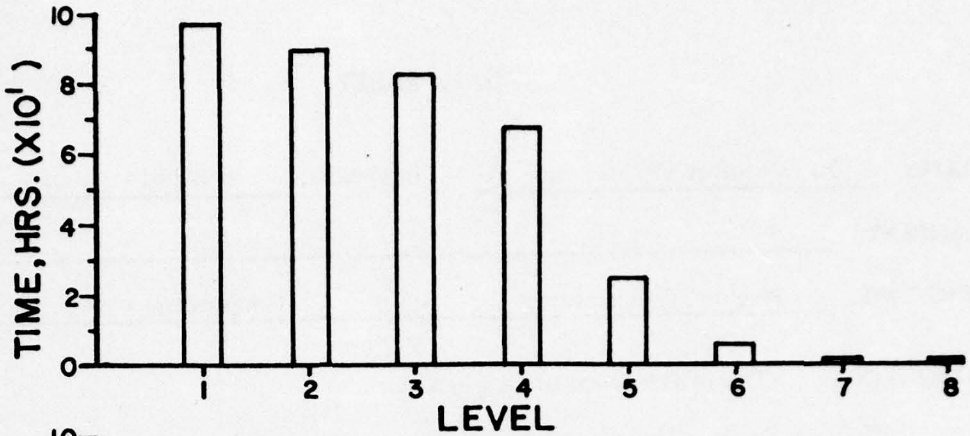
Off Road Truck – Hydraulic Lift System, Southwest U.S.

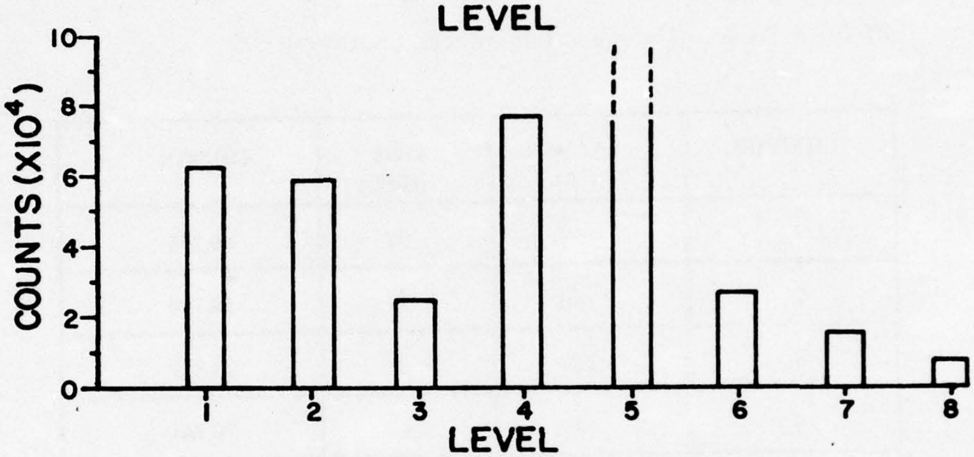
CHANNEL	LEVEL (°F)	TIME (HRS.)	COUNTS
1	85	97	60,234
2	106	89	59,740
3	127	83	23,827
4	147	68	76,361
5	168	24	> 75,000
6	189	3.7	24,177
7	209	.4	10,449
8	230	.4	6,304

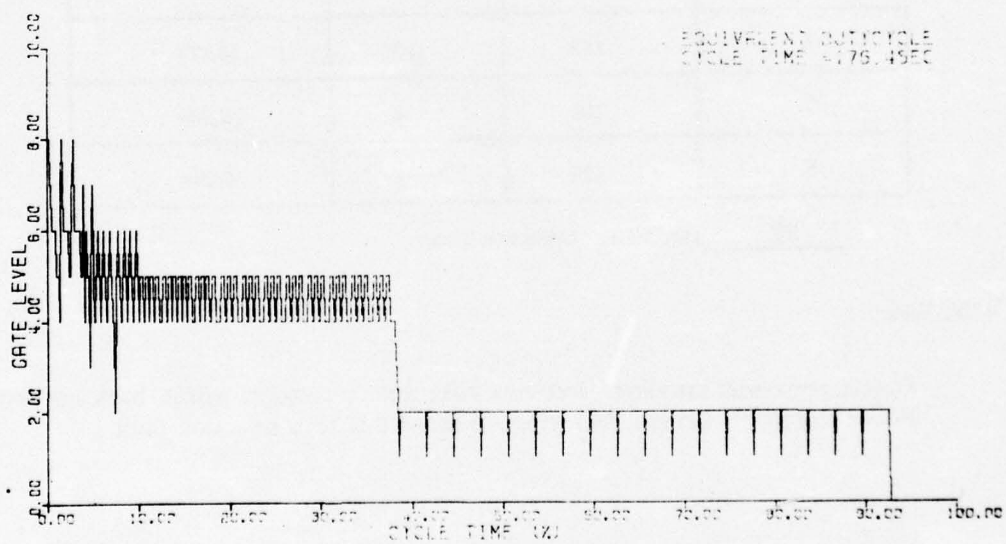
98 (Hrs.) Total Operation Time

REMARKS:

Project personnel installed. Extreme EM1 due to solenoid valves, back up horn. Filters installed. BD No. 141 pressure failed due to connection fault.


TIME, HRS. (X10¹)
10
8
6
4
2
0
1
2
3
4
5
6
7
8
LEVEL


COUNTS (X10⁴)
10
8
6
4
2
0
1
2
3
4
5
6
7
8
LEVEL


EQUIVALENT DUTYCYCLE
CYCLE TIME =176.4SEC
GATE LEVEL
10.00
8.00
6.00
4.00
2.00
0.00
0.00
10.00
20.00
30.00
40.00
50.00
60.00
70.00
80.00
90.00
100.00
CYCLE TIME (%)

DATA SHEET

DATE: ______________________ **UNIT NO.:** 1019

COMPANY: 5

UNIT TYPE: Pressure

APPLICATION: *(Type of Vehicle and Location of Sensors)*

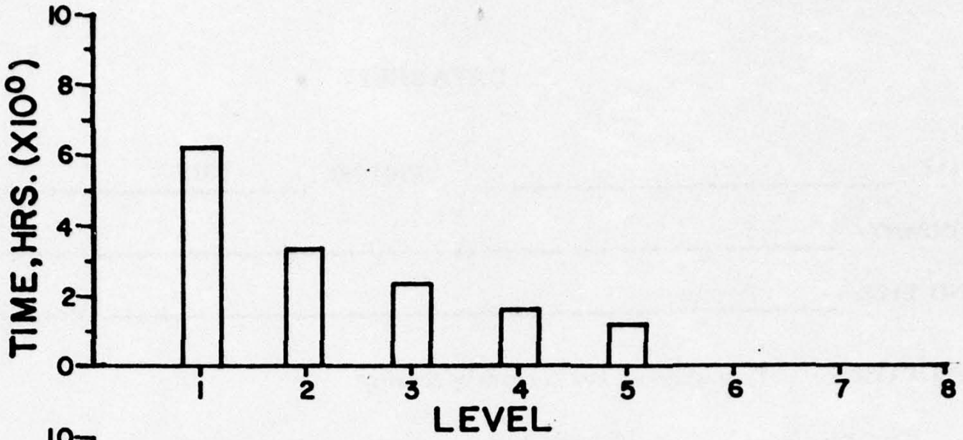
Scraper, Main System, Midwest U.S.

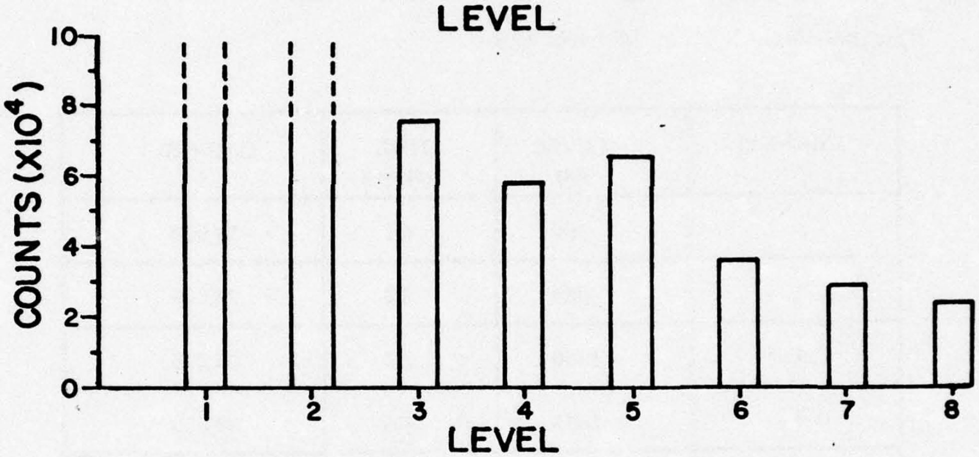
CHANNEL	LEVEL PSI	TIME (HRS.)	COUNTS
1	500	6.1	> 75,000
2	885	3.2	> 75,000
3	1250	2.2	75,000
4	1675	1.6	58,000
5	2050	1.1	64,000
6	2420	0.0	36,000
7	2850	0.0	27,000
8	3200	0.0	22,000

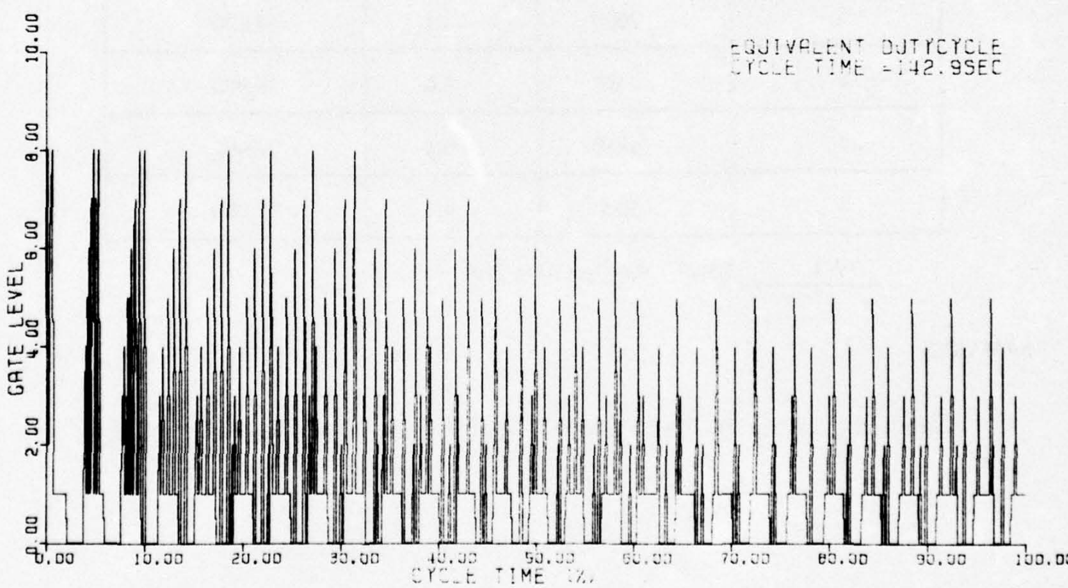
79.4 **(Hrs.) Total Operation Time**

REMARKS:

Data indicates severe EM1. Sponsor testing discontinued on this vehicle.


TIME, HRS. (X10⁰)
LEVEL
0
2
4
6
8
10
1
2
3
4
5
6
7
8


COUNTS (X10⁴)
LEVEL
0
2
4
6
8
10
1
2
3
4
5
6
7
8


EQUIVALENT DUTYCYCLE
CYCLE TIME =142.9SEC
GATE LEVEL
CYCLE TIME (%)
0.00
2.00
4.00
6.00
8.00
10.00
0.00
10.00
20.00
30.00
40.00
50.00
60.00
70.00
80.00
90.00
100.00

DATA SHEET

DATE: 31 October 1975 **UNIT NO.:** 1003

COMPANY: 6

UNIT TYPE: Pressure

APPLICATION: *(Type of Vehicle and Location of Sensors)*

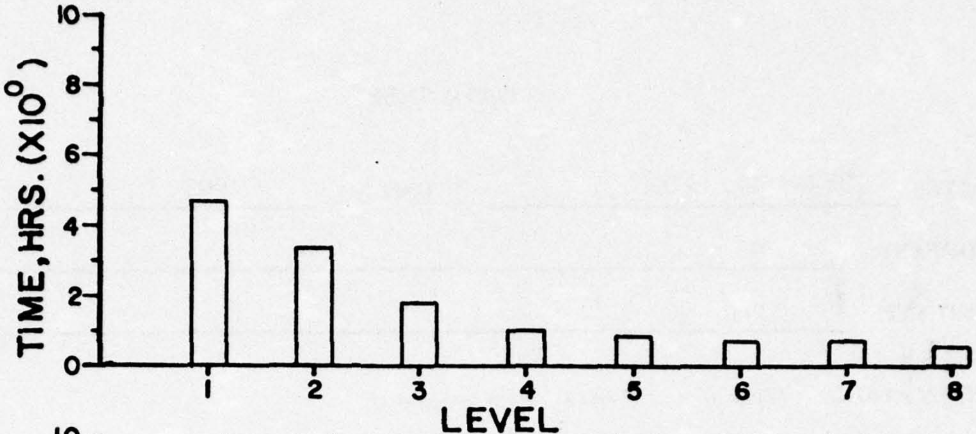
Front End Loader – Main System, Midwest U.S.

CHANNEL	LEVEL PSI	TIME (HRS.)	COUNTS
1	500	4.6	> 75,000
2	925	3.4	
3	1350	1.8	
4	1775	1.0	
5	2200	.9	
6	2625	.7	
7	3050	.7	
8	3475	.6	

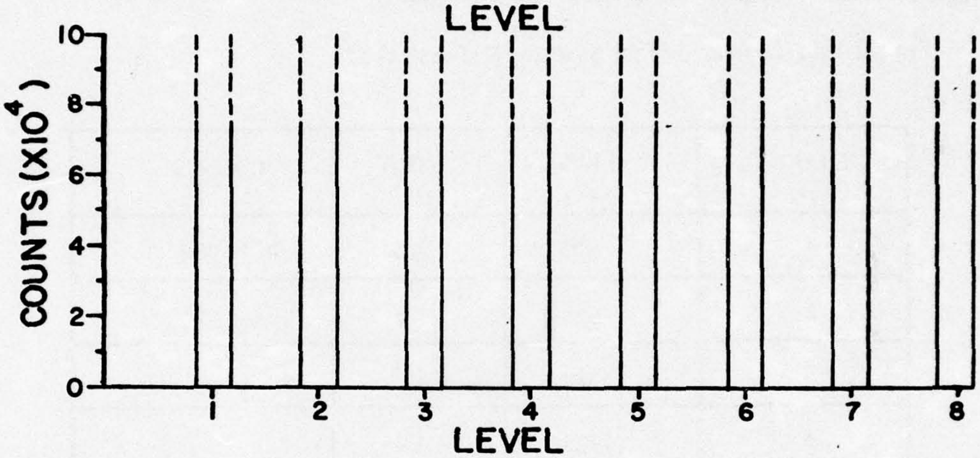
23.7 **(Hrs.) Total Operation Time**

REMARKS:

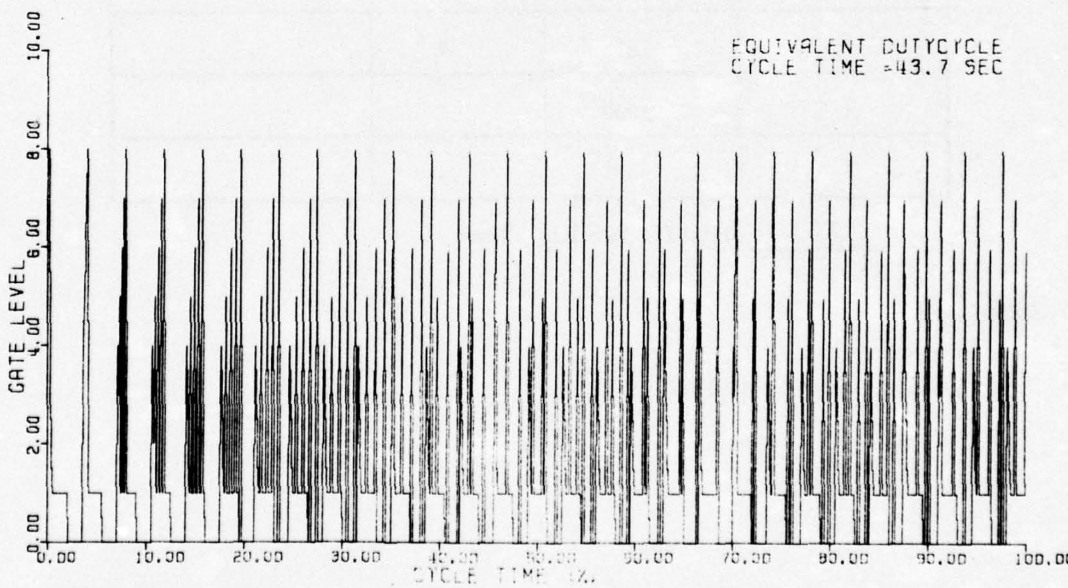
Data shows extreme EM1 problem.



TIME, HRS. (X10^0)
0
2
4
6
8
10
1 2 3 4 5 6 7 8
LEVEL



COUNTS (X10^4)
0
2
4
6
8
10
1 2 3 4 5 6 7 8
LEVEL



EQUIVALENT DUTYCYCLE
CYCLE TIME =43.7 SEC
GATE LEVEL
0.00
2.00
4.00
6.00
8.00
10.00
0.00 10.00 20.00 30.00 40.00 50.00 60.00 70.00 80.00 90.00 100.00
CYCLE TIME (%)

DATA SHEET

DATE: 30 October 1975 **UNIT NO.:** 1068–167

COMPANY: 2

UNIT TYPE: Strain (Experimental Model)

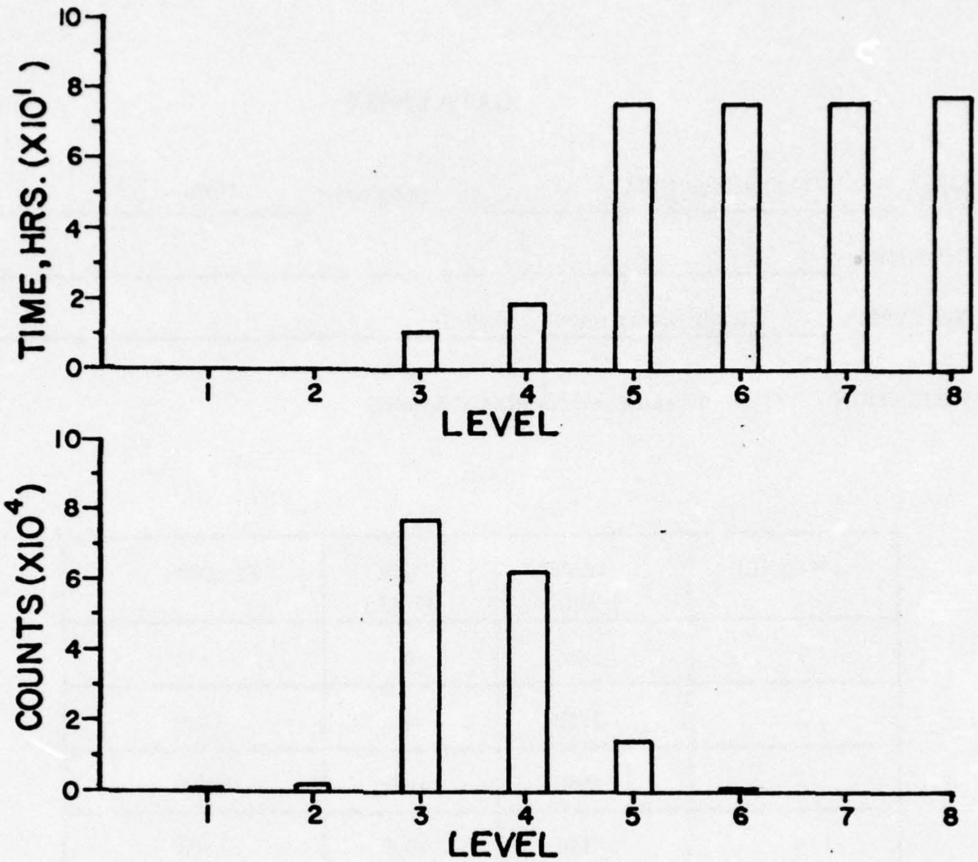
APPLICATION: *(Type of Vehicle and Location of Sensors)*

CHANNEL	LEVEL μ in/in	TIME (HRS.)	COUNTS
1	–2450	0	130
2	–1750	0	1340
3	–1050	10.0	> 75,000
4	– 350	19.5	61,700
5	+ 350	75.3	12,311
6	+1050	75.3	276
7	+1750	75.24	0
8	+2450	76.03	0

99.2 **(Hrs.) Total Operation Time**

REMARKS:

Structure failed – data interpretation difficult due to yielding which resulted in sustained offset.



Equivalent duty cycle not available.

DATA SHEET

DATE: 8 October 1975 **UNIT NO.:** 1060–159

COMPANY: 7

UNIT TYPE: Pressure/Temperature Temperature = 159

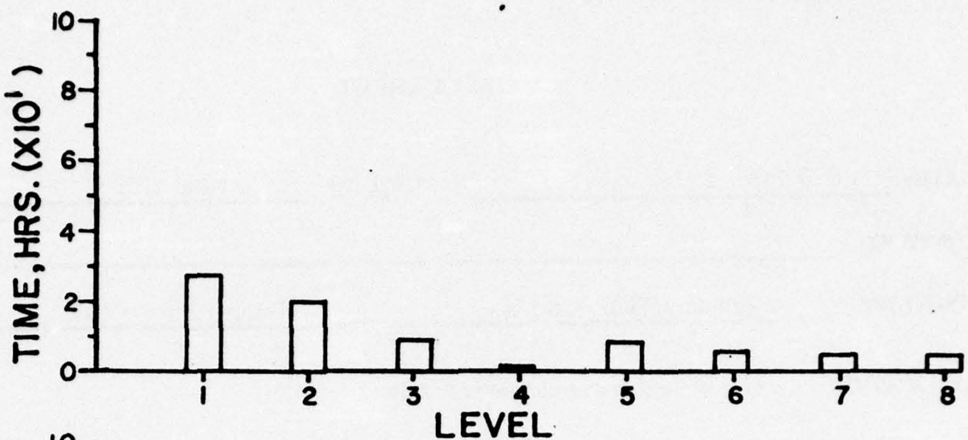
APPLICATION: *(Type of Vehicle and Location of Sensors)*

Wheeled Tractor, Main System, Midwest U.S.

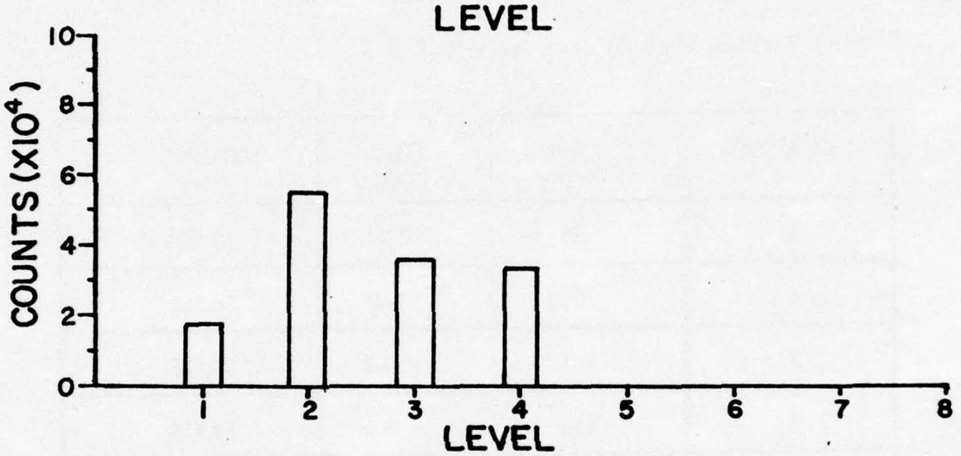
CHANNEL	LEVEL (°F)	TIME (HRS.)	COUNTS
1	85	27.4	17,259
2	106	20.9	54,383
3	128	9.7	35,825
4	150	3.3	33,928
5	170	.9	0
6	192	.7	0
7	213	.5	0
8	234	.5	0

29.2 **(Hrs.) Total Operation Time**

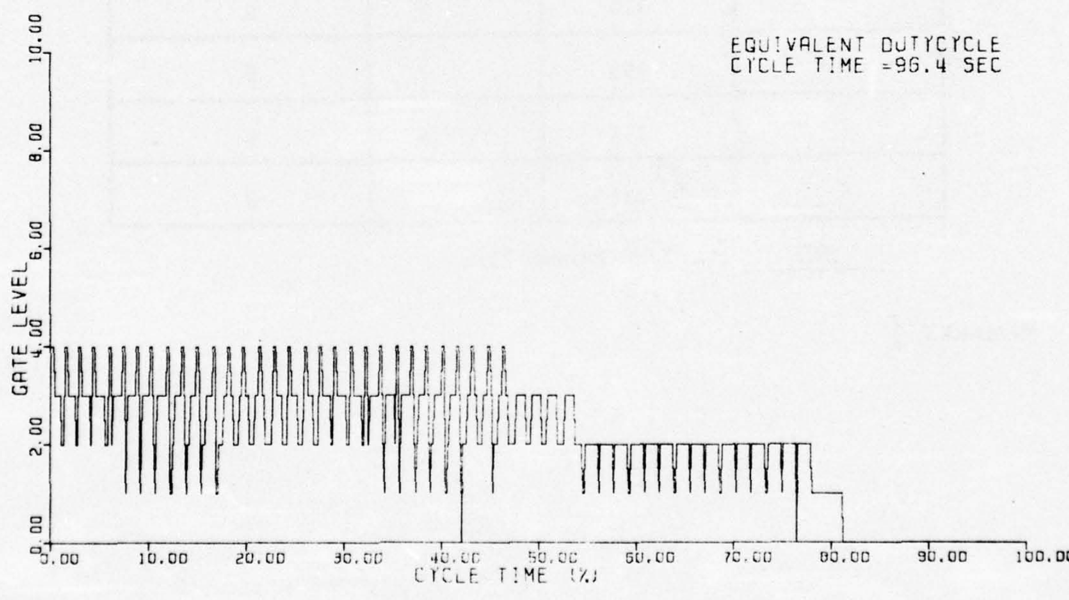
REMARKS:



TIME, HRS. (X10^{1})
LEVEL



COUNTS (X10^{4})
LEVEL



EQUIVALENT DUTYCYCLE
CYCLE TIME =96.4 SEC
GATE LEVEL
CYCLE TIME (%)

DATA SHEET

DATE: 8 October 1975 **UNIT NO.:** 1060–160

COMPANY: 7

UNIT TYPE: Pressure/Temperature Pressure = 160

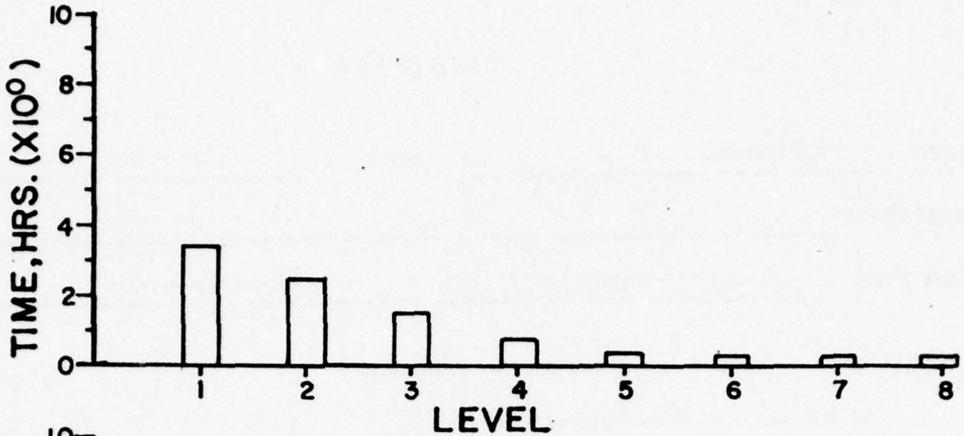
APPLICATION: *(Type of Vehicle and Location of Sensors)*

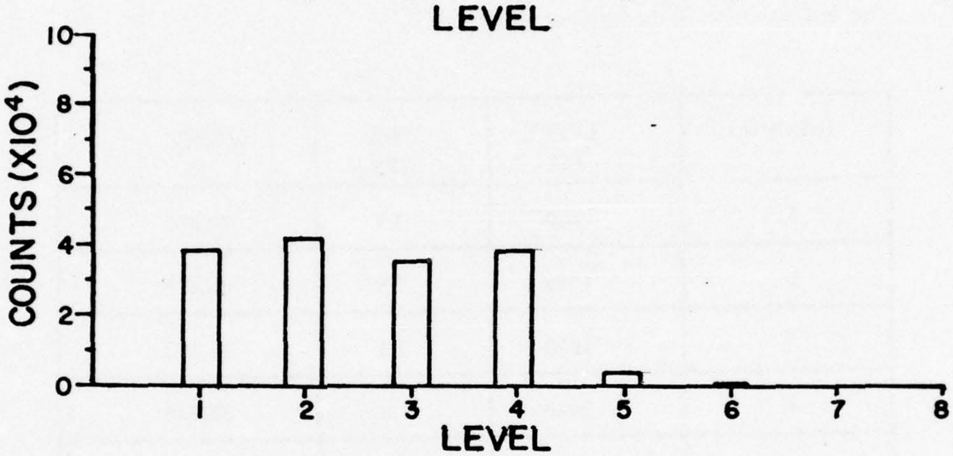
Wheeled Tractor, Main System, Midwest U.S.

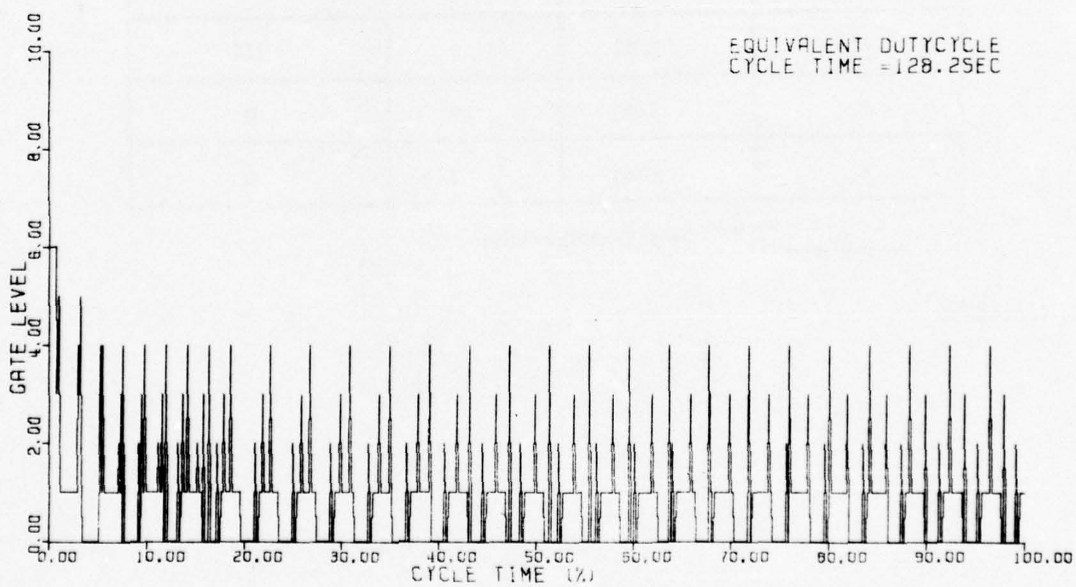
CHANNEL	LEVEL PSI	TIME (HRS.)	COUNTS
1	620	3.3	38,095
2	1090	2.5	40,873
3	1570	1.5	35,793
4	2040	.8	38,545
5	2510	.3	2,570
6	2980	.2	222
7	3470	.2	0
8	3940	.2	0

29.2 **(Hrs.) Total Operation Time**

REMARKS:


TIME, HRS. (X10^0)
LEVEL


COUNTS (X10^4)
LEVEL


EQUIVALENT DUTYCYCLE
CYCLE TIME =128.2SEC
GATE LEVEL
CYCLE TIME (%)

DATA SHEET

DATE: 20 October 1975 **UNIT NO.:** 1009

COMPANY: 7

UNIT TYPE: Pressure

APPLICATION: *(Type of Vehicle and Location of Sensors)*

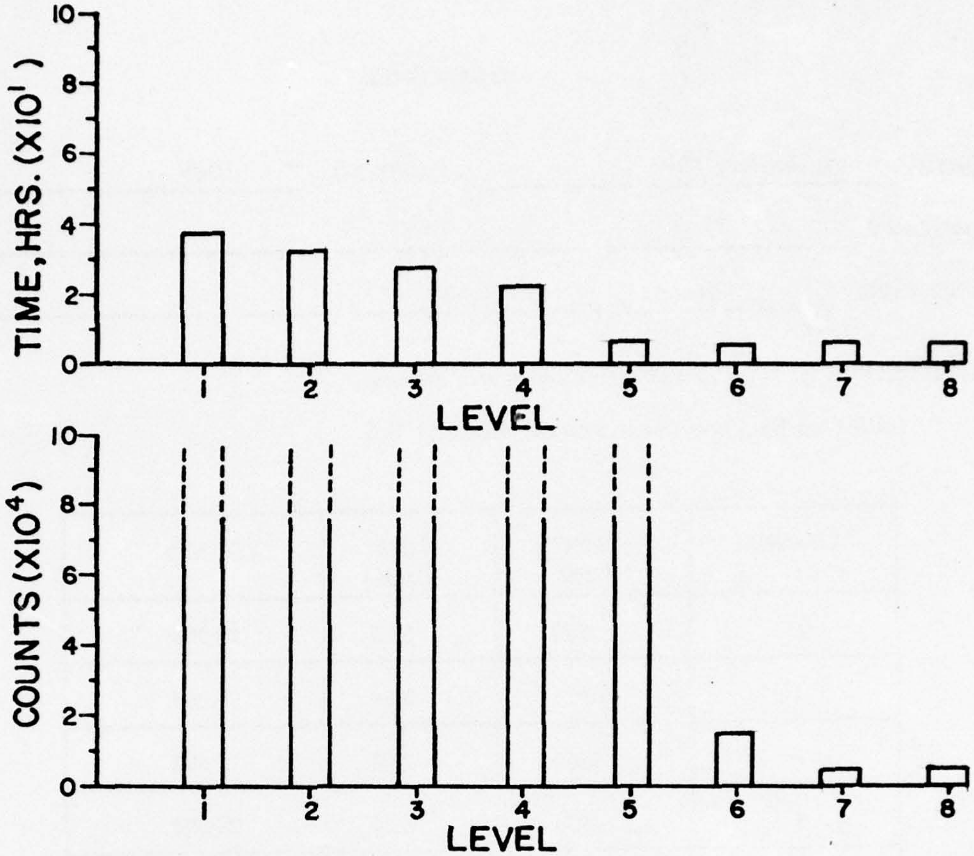
Industrial Backhoe, Main System, Midwest U.S.

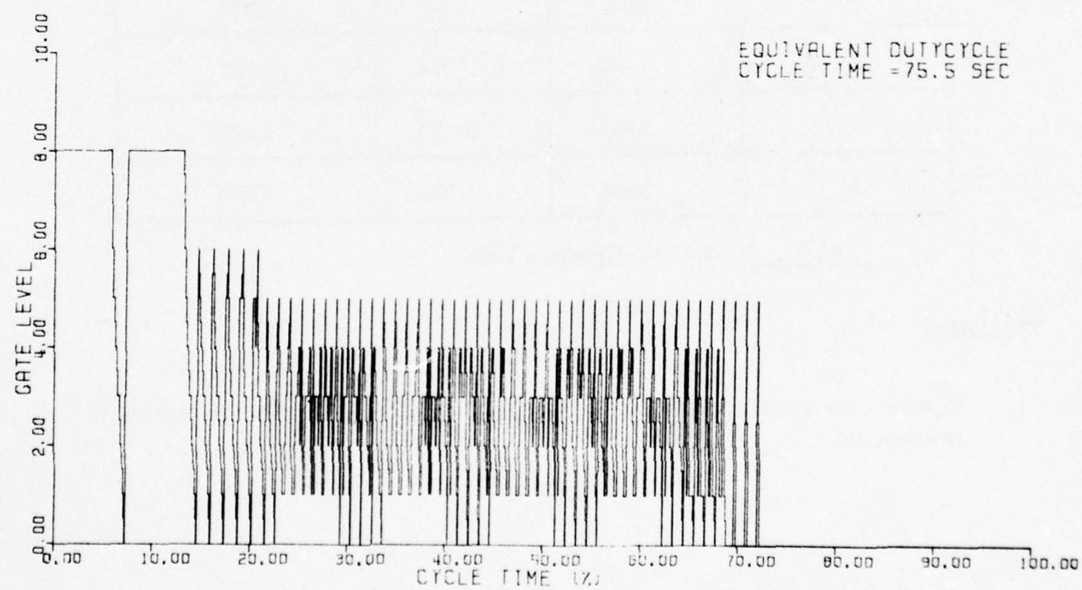
CHANNEL	LEVEL PSI	TIME (HRS.)	COUNTS
1	500	37.6	> 75,000
2	975	32.4	> 75,000
3	1450	27.4	> 75,000
4	1925	21.4	> 75,000
5	2400	5.8	> 75,000
6	2900	5.1	14,900
7	3350	5.0	3,400
8	3850	5.0	3,600

41.4 **(Hrs.) Total Operation Time**

REMARKS:

Open-center system with relief set at 2400 psi. Pump ripple characteristics not known.


TIME, HRS. (X10^1)
LEVEL
COUNTS (X10^4)
LEVEL


EQUIVALENT DUTYCYCLE
CYCLE TIME =75.5 SEC
GATE LEVEL
CYCLE TIME (%)

DATA SHEET

DATE: 20 October 1975 UNIT NO.: 1008

COMPANY: 7

UNIT TYPE: Temperature

APPLICATION: *(Type of Vehicle and Location of Sensors)*

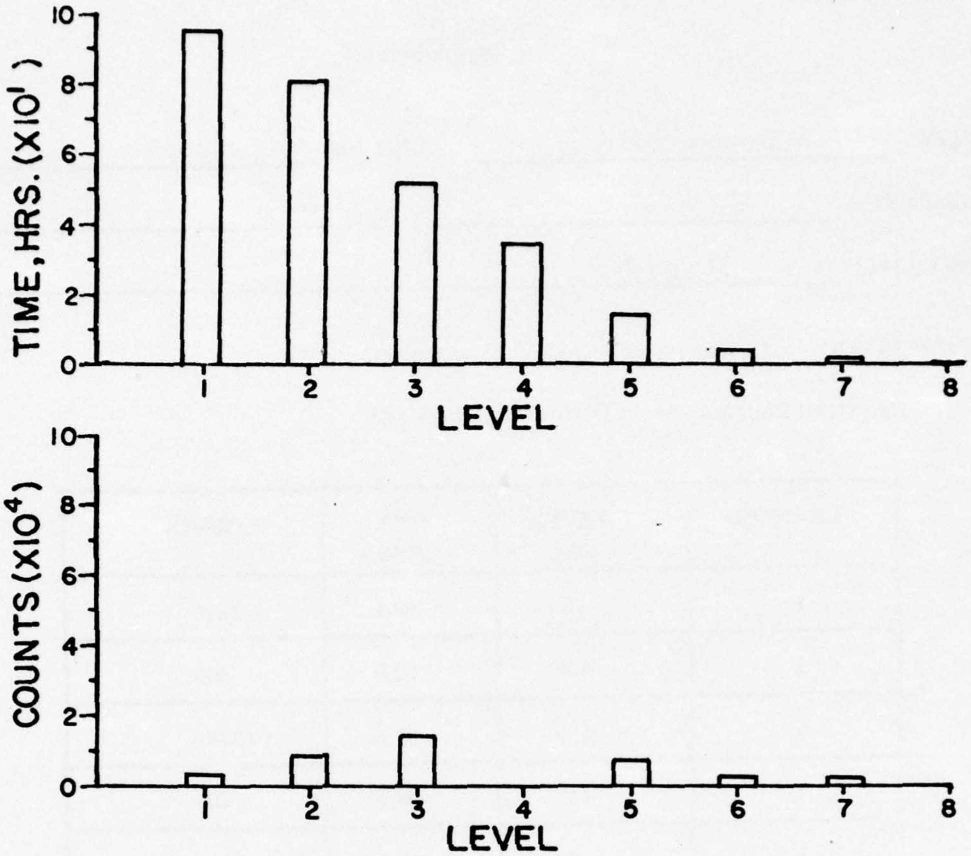
Industrial Backhoe, Main System, Midwest U.S.

CHANNEL	LEVEL (°F)	TIME (HRS.)	COUNTS
1	84	94.1	3426
2	108	80.0	8402
3	130	51.6	13,694
4	152	34.0	0*
5	174	13.7	7891
6	197	3.6	3295
7	220	.7	3111
8	241	.1	0

99.9 **(Hrs.) Total Operation Time**

REMARKS:

*No fault detected.



Equivalent duty cycle not available.

DATA SHEET

DATE: 15 April 1976 **UNIT NO.:** 1008

COMPANY: 7

UNIT TYPE: Temperature

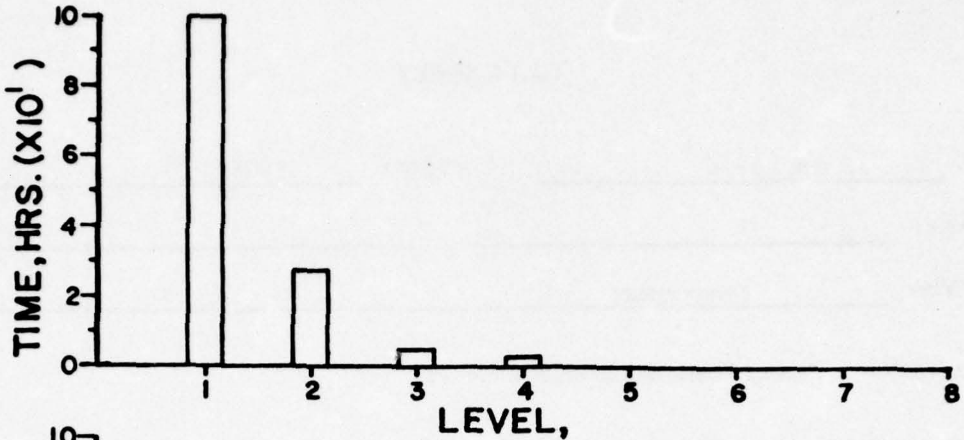
APPLICATION: *(Type of Vehicle and Location of Sensors)*

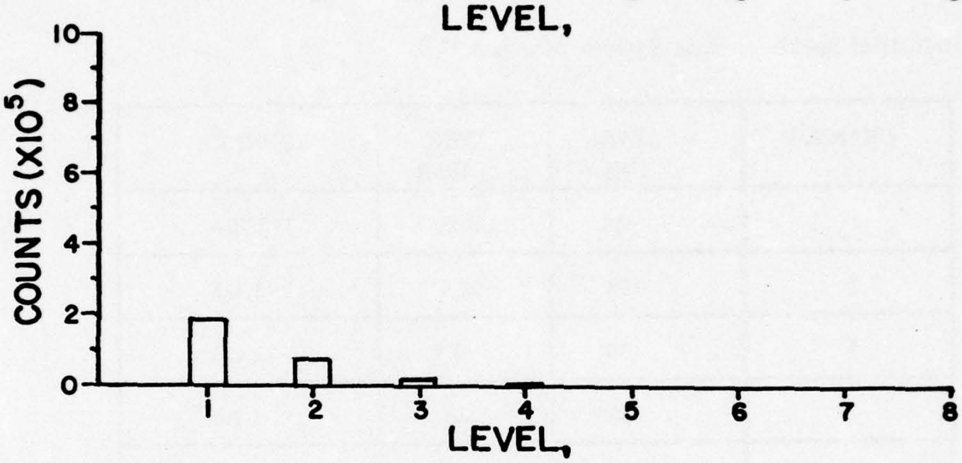
Industrial Backhoe, Main System, Midwest U.S.

CHANNEL	LEVEL (°F)	TIME (HRS.)	COUNTS
1	84	100.0	118,700
2	108	27.9	93,425
3	130	3.6	21,492
4	152	2.1	1,205
5	174	0	0
6	197	0	0
7	220	0	0
8	241	0	0

100 **(Hrs.) Total Operation Time**

REMARKS:





Equivalent duty cycle not available.

DATA SHEET

DATE: 4 September 1975 **UNIT NO.:** 1232A

COMPANY: 7

UNIT TYPE: Pressure

APPLICATION: *(Type of Vehicle and Location of Sensors)*

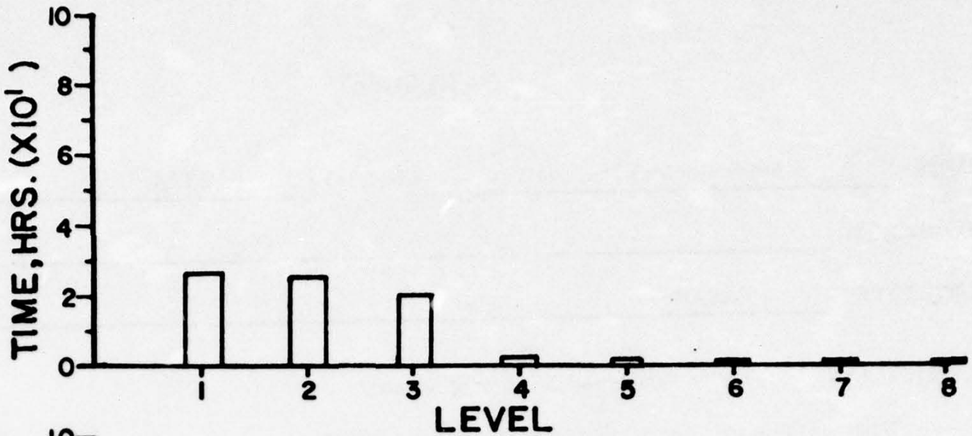
Wheeled Tractor, Transmission Return Line Filter

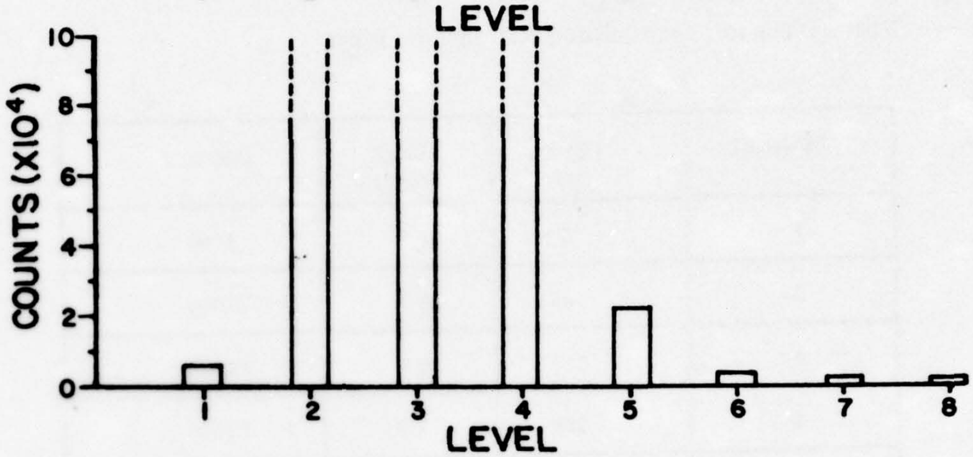
CHANNEL	LEVEL PSI	TIME (HRS.)	COUNTS
1	72	26.0	5760
2	145	25.2	> 75,000
3	217	20.1	> 75,000
4	289	1.0	> 75,000
5	362	0.6	22,100
6	436	0.5	3950
7	509	0.5	1860
8	583	0.4	1580

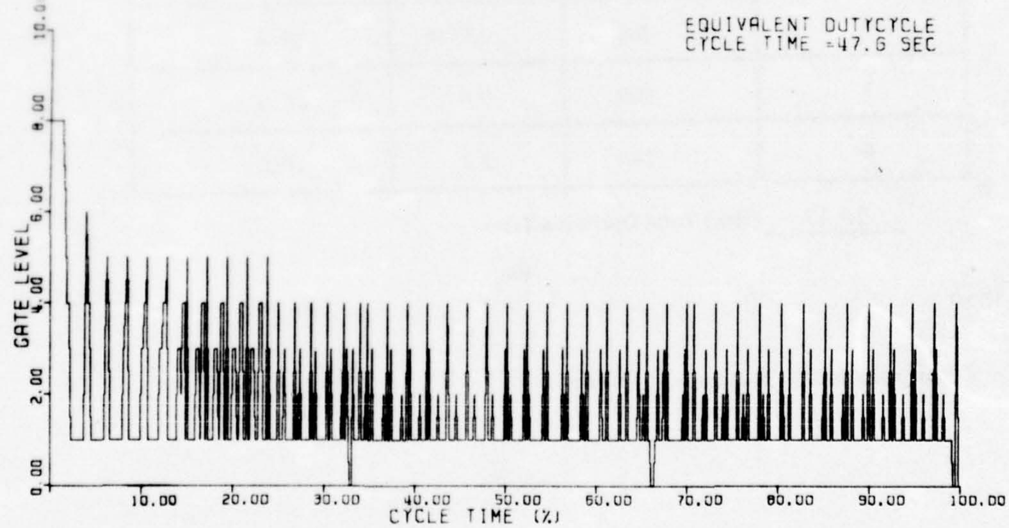
26.47 **(Hrs.) Total Operation Time**

REMARKS:

Filter collapsed. Transmission failed.


TIME, HRS. (X10¹)
LEVEL


COUNTS (X10⁴)
LEVEL


EQUIVALENT DUTYCYCLE
CYCLE TIME =47.6 SEC
GATE LEVEL
CYCLE TIME (%)

DATA SHEET

DATE: 4 September 1975 **UNIT NO.:** 1232B

COMPANY: 7

UNIT TYPE: Pressure

APPLICATION: *(Type of Vehicle and Location of Sensors)*

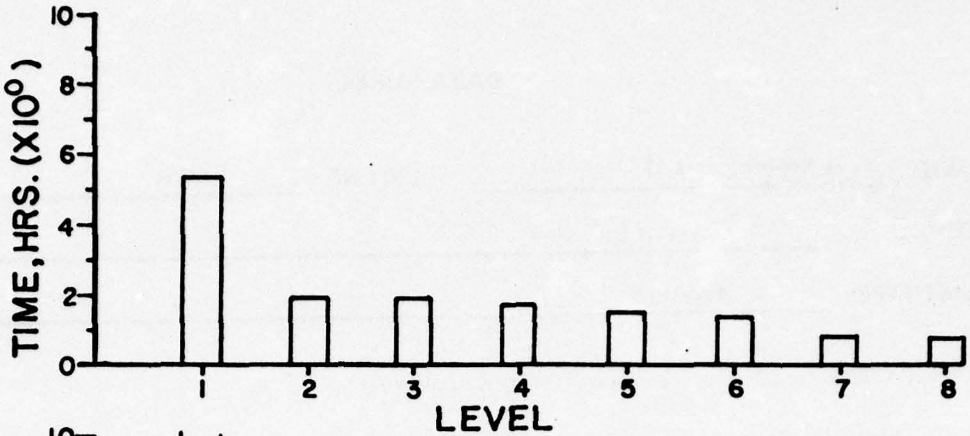
Wheeled Tractor, Transmission Shift Pressure

CHANNEL	LEVEL PSI	TIME (HRS.)	COUNTS
1	683	5.3	> 75,000
2	1383	2.0	7570
3	2100	2.0	2460
4	2817	1.8	745
5	3533	1.4	718
6	4250	1.3	472
7	4967	.8	268
8	5683	.8	239

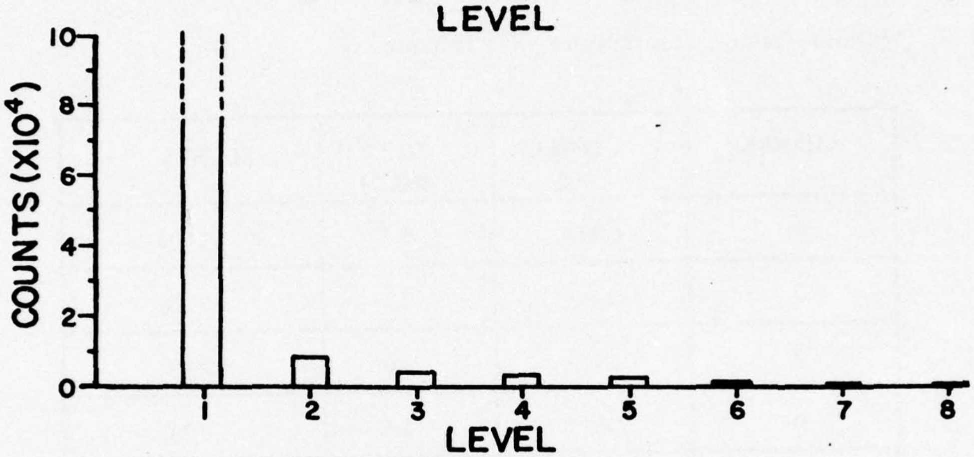
26.47 **(Hrs.) Total Operation Time**

REMARKS:

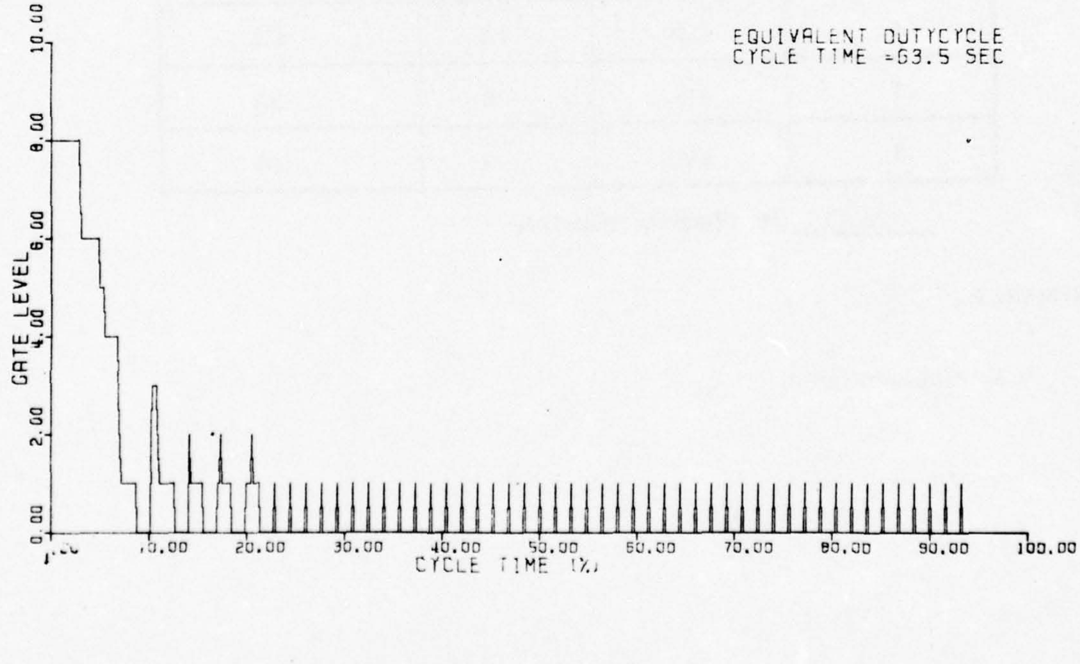
Transmission failed.



TIME, HRS. (X10⁰)
LEVEL



COUNTS (X10⁴)
LEVEL



EQUIVALENT DUTYCYCLE
CYCLE TIME =63.5 SEC
GATE LEVEL
CYCLE TIME (%)

DATA SHEET

DATE: 15 March 1976 **UNIT NO.:** 1007

COMPANY: 8

UNIT TYPE: Temperature

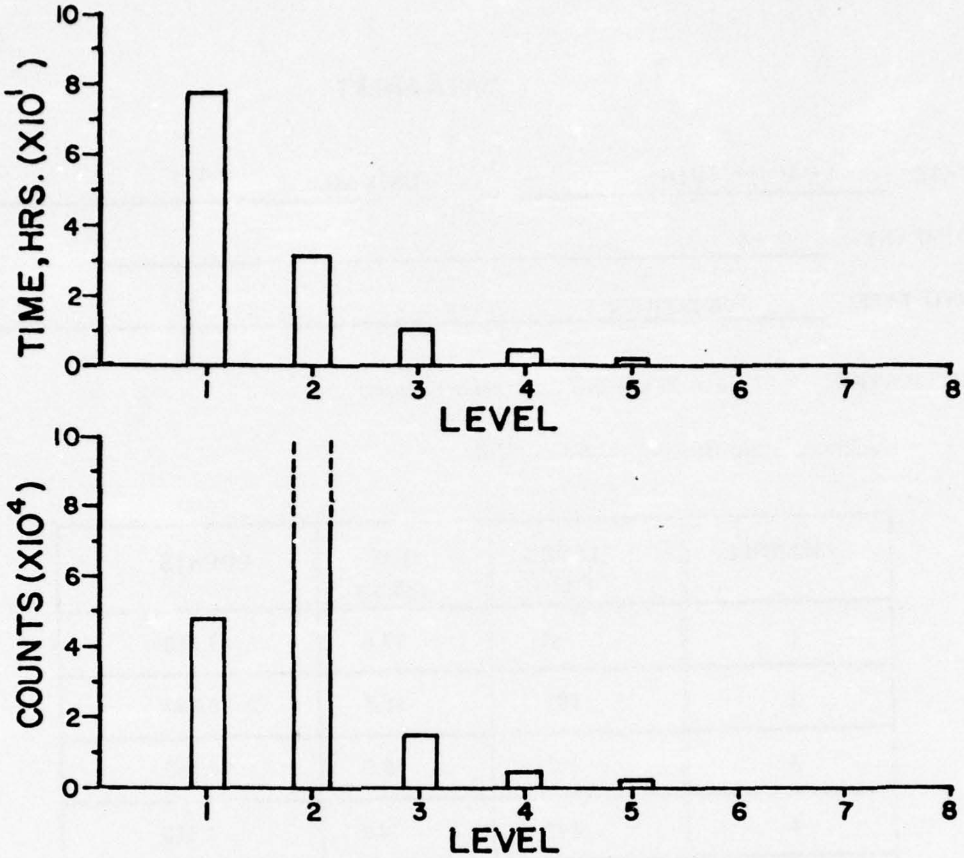
APPLICATION: *(Type of Vehicle and Location of Sensors)*

Backhoe, Main System, Midwest U.S.

CHANNEL	LEVEL (°F)	TIME (HRS.)	COUNTS
1	81	77.6	47,510
2	101	31.8	> 75,000
3	122	10.5	14,380
4	142	4.7	2,610
5	162	2.9	1,900
6	183	0	0
7	201	0	0
8	224	0	0

100 **(Hrs.) Total Operation Time**

REMARKS:



Equivalent duty cycles not available.

DATA SHEET

DATE: 15 September 1974 **UNIT NO.:** 1004

COMPANY: 9

UNIT TYPE: Temperature

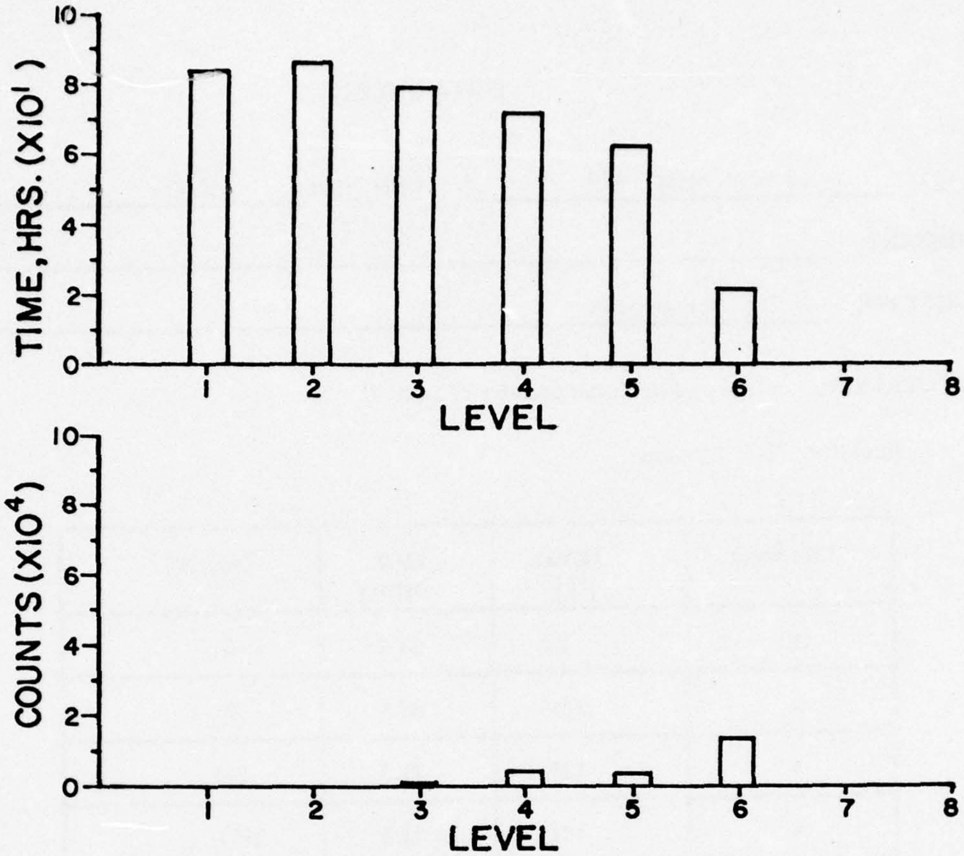
APPLICATION: *(Type of Vehicle and Location of Sensors)*

Backhoe, Main System

CHANNEL	LEVEL (°F)	TIME (HRS.)	COUNTS
1	84	83.4	0
2	106	85.8	0
3	129	79.3	143
4	151	71.6	2917
5	173	62.1	2832
6	195	21.8	11,834
7	217	0	0
8	239	0	0

87.9 **(Hrs.) Total Operation Time**

REMARKS:



Equivalent duty cycles not available.

DATA SHEET

DATE: 25 April 1975 UNIT NO.: 1017

COMPANY: 2

UNIT TYPE: Pressure

APPLICATION: *(Type of Vehicle and Location of Sensors)*

Industrial Tractor, Main System

CHANNEL	LEVEL PSI	TIME (HRS.)	COUNTS
1	544	10.4	57,012
2	943	4.1	> 75,000
3	1333	3.8	> 75,000
4	1723	2.2	58,261
5	2122	0.5	19,366
6	2512	0.4	1,676
7	2912	0.4	196
8	3311	0.3	600

__________ (Hrs.) Total Operation Time

REMARKS:

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SECTION VI

PUMP CONTAMINANT TOLERANCE VERIFICATION

PROJECT STAFF

L. E. Bensch, Program Manager

L. V. Grade, Project Associate

L. C. Moore, Project Associate

FOREWORD

This section presents a detailed account of the project activities in the area of pump contaminant tolerance verification. A brief summary of pump contaminant sensitivity test procedures and a mathematical interpretation technique are given. The primary objective of this project was to verify this life prediction theory for various contaminated conditions. Results of several extended life tests are presented and compared to the calculated values.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

CHAPTER I

INTRODUCTION

Since the hydraulic pump is the prime energy source for a fluid power system, the importance of protecting this element cannot be overstressed. The pump is generally recognized as one of the most sensitive of all hydraulic components to contaminant wear; therefore, a knowledge of its tolerance and operating conditions is critical for proper contamination control. This report presents the results of an initial study conducted to verify existing pump contaminant sensitivity tests and wear prediction methods.

A great deal of effort has been expended during the past several years by the Fluid Power Research Center at Oklahoma State University, much under U.S. Army Mobility Equipment Research and Development Command sponsorship, relative to proper hydraulic system contamination control. The primary objective of a complete contamination control appraisal is to estimate the service life of a hydraulic system under the presence of contamination. This is called the contaminant service life and is evaluated explicitly in terms of acceptable performance. It is generally accepted that the contaminant service life of a hydraulic component is a function of the contamination level of the system fluid and the contaminant sensitivity of the particular component. The contamination level of the fluid is determined by the particulate separation capability of the system filter and the amount and distribution of particulate contaminant which enters the system. The contaminant sensitivity of a component (e.g., hydraulic pump) is a function of the design parameters and operating conditions – pressure, speed, fluid, and temperature. A complete contamination control appraisal includes consideration of all of these influencing parameters.

Under the sponsorship of the U.S. Army MERDC, a test procedure was developed at the Fluid Power Research Center for evaluating the sensitivity of hydraulic pumps to contaminant wear. This procedure, which measures the flow degradation of a pump under the presence of contaminant, is discussed in Chapter II. To complement the pump contaminant sensitivity procedure, a separate but related study was conducted for the Basic Fluid Power Research Program at Oklahoma State University to develop a contaminant wear theory for pumps which could be utilized to calculate the filter protection required to obtain adequate field life. This theory, which is briefly reviewed in Chapter III, has been completed and documented; however, no verification program has been conducted to establish its utility. In order that the test procedure and wear theory could be utilized for determining proper filtration levels for MERDC systems, the study reported herein was conducted to verify the relationships developed. Long-term contaminant life tests were conducted on pumps which were also evaluated in accordance with the existing standard contaminant sensitivity tests. These tests and results are reported in Chapter IV. A discussion of the results along with conclusions relative to this study is given in Chapter V. The actual test data are included in the appendices.

CHAPTER II

PUMP CONTAMINANT SENSITIVITY TEST

The contaminant tolerance of a hydraulic pump is determined by conducting a special contaminant sensitivity test. The test procedure with some later modifications was formalized at the Fluid Power Research Center under the sponsorship of the U.S. Army MERDC [1]. The test is designed to progressively expose a pump to increasing sizes of contaminant while monitoring the influence of each size on the designated performance parameter – output flow. The pump is operated at reference or rated conditions of shaft speed and outlet pressure throughout the test.

The test facility utilized to conduct a contaminant sensitivity test is illustrated in Fig. 2-1. To enhance both the repeatability and reproducibility of the test, the volume of the test fluid in the circuit is maintained at a constant value equal to one-fourth the rated volume flow of the pump per minute. The control or clean-up filters utilized must be capable of reducing the contamination level of the fluid between injections to less than 10 mg/litre. Again, for the sake of uniformity, the procedure specifies the use of single-cut (e.g., 0-5, 0-10, 0-20μM, etc.) contaminant classified from AC Fine Test Dust as the base stock. A quantity of contaminant in each size range is progressively introduced to yield a test contamination level of 300 mg/litre.

Prior to injecting contaminant into the test system, a break-in or stabilization period must be established. During this period, the test pump is operated at progressively higher pressures until the rated pressure is reached. This pressure is maintained for at least one hour until the output flow stabilizes. This break-in strengthens the contention that any subsequent wear is due to contamination.

The contaminant sensitivity test is initiated by injecting the required quantity of 0-5 micrometre dust into the test loop. The contaminant is circulated under constant operating conditions until 30 minutes have elapsed or until the output flow has remained constant for at least 10 minutes. After the required contaminant exposure, the fluid is circulated through the control filter until the desired fluid cleanliness is achieved. At the end of the filter period, the amount of degradation in output flow resulting from the previous injection is measured and recorded. The above test sequence is continued with progressively increasing particle size ranges – 0-10, 0-20, ... 0-80μM, until either the 0-80μM size has been injected or the output flow has decreased to less than 70% of its original value. The data resulting from this test can then be utilized to calculate contaminant tolerance information.

The pump contaminant sensitivity test procedure developed at OSU was submitted to the National Fluid Power Association for industrial standardization. It has undergone some modifications during the past few years and now appears in NFPA as *"Method for Establishing the Flow Degradation of Hydraulic Fluid Power Pumps When Exposed to Particulate Contaminant"* [2]. The latest draft was recently balloted, and NFPA approval as a national standard is anticipated in the near future.

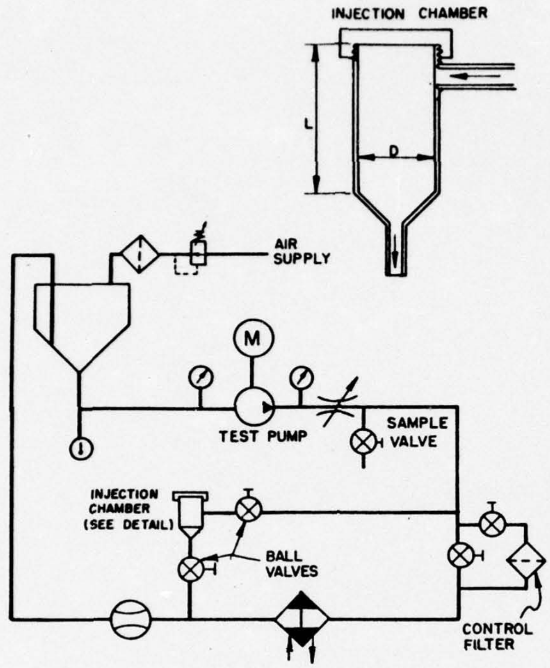

Fig. 2-1. Test Circuit for Pump Contaminant Sensitivity Test.

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CHAPTER III

CONTAMINANT WEAR THEORY

The primary requirement for a contaminant wear theory or sensitivity model is to provide a means by which the effects of changes in operating environment upon component service life can be determined. Since a contaminant sensitivity test is conducted with 300 mg/litre of specially sized contaminants, an appropriate model must be utilized for prediction of wear under other contaminant environments. The following contaminant wear theory which was developed at Oklahoma State University and presented in Ref. [3] is summarized here to provide the reader with an adequate background for understanding and applying the model.

LABORATORY SENSITIVITY EXPRESSIONS

The contaminant sensitivity model is based upon the contention that, for every critical size particle that passes through or is exposed to the pump, there is a finite amount of damage which reduces the output flow of the pump. Thus, the rate that the flow degrades (dQ/dt) depends upon the sensitivity (S_i) of the pump at size interval (i) and the rate (dN_i/dt) at which particles of size interval (i) are exposed to the pump. This relationship is expressed by the following expression:

$$dQ/dt = -S_i\,(dN_i/dt) \qquad (3\text{-}1)$$

The contaminant wear equation can be evaluated for conditions of the laboratory test by considering the nature of each parameter. The rate at which particles of any size interval are exposed to the internal parts of a pump at time t with flow rate Q and particle concentration n is determined by:

$$dN/dt = Qn \tag{3-2}$$

Higher flow rates or particle concentrations result in higher exposure rates.

A factor which has a different characteristic in the laboratory test than it does in a field application is the particle concentration. In the field application, the particle concentration is maintained relatively constant by the continual filtering and replenishment from the environment. This situation does not exist in the laboratory test, since, for a given size range, particles are only injected once and are eventually destroyed by the test pump. The particle destruction process is reflected by:

$$n = n_o e^{-t/\tau} \tag{3-3}$$

where n_o is the initial particle concentration and τ is the time constant of the destruction process.

It has been demonstrated [4] that the contaminant sensitivity is a linear function of the particle concentration. Thus, the sensitivity can be expressed as:

$$S = \alpha n \tag{3-4}$$

where α is a constant termed the contaminant wear coefficient and has units of (volume/particle)^2 per unit time.

Combining Eqs. (3-1) through (3-4) yields the differential equation:

$$dQ/dt = -\alpha n_o^2 Q e^{-2t/\tau} \tag{3-5}$$

Integrating Eq. (3-5) yields the governing equation for the laboratory wear process:

$$Q = Q_o e^{-\alpha \tau n_o^2 (1 - e^{-2t/\tau})} \tag{3-6}$$

where Q_o is the initial flow rate prior to injection and Q is the flow rate at time t.

If the contaminant wear coefficient α and the particle destruction time constant τ for one particle size interval are known, Eq. (3-6) can be utilized to predict the flow rate of the pump at any time after the injection. Conversely, if the flow rate versus time curve is known, Eq. (3-6) can be utilized to solve for α and τ for each particle size interval. It was shown in Ref. [3] that the particle destruction time constant was equal to an average value of approximately nine minutes for all particle sizes and types of pumps. If it is assumed that τ is a constant, then Eq. (3-6) can be rewritten for a given particle size interval and $t = \infty$ as:

$$\alpha = -2 \ln (Q_f/Q_o) \;/\; n_o^2 \tag{3-7}$$

where Q_f is the final flow rate after a given injection.

The test data necessary to evaluate the wear coefficients for each particle size are obtained directly from the laboratory tests. Typical data which can be observed during the test are illustrated in Fig. 3-1 and show the normal influence of particle size on the flow degradation of the pump. The contaminant wear coefficients are characteristic constants for a given pump and can be utilized to predict the wear associated with the identical pumps under different contaminant environments and similar operating conditions.

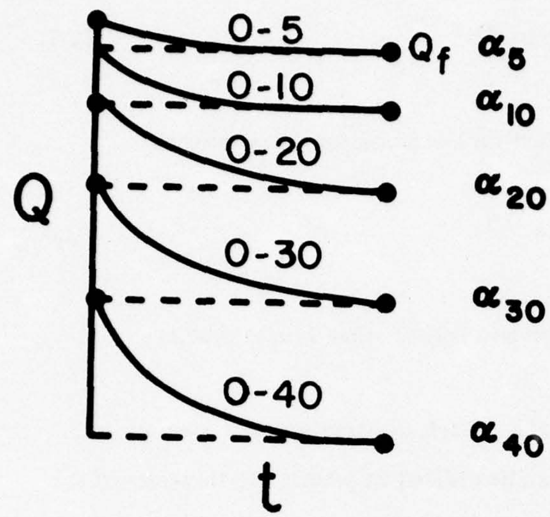

Fig. 3-1. Size Sensitivity Relationship.

FIELD SENSITIVITY EXPRESSIONS

For field operation, it is assumed that the contaminant is continually being ingressed or generated and subsequently filtered from a system. This continuous interchange of new particles results in a relatively constant contamination level. Thus, the governing differential equation derived similar to Eq. (3-5) is:

$$dQ/dt = -\alpha n^2 Q \qquad (3\text{-}8)$$

for a given particle size interval and constant particle concentration n. Integration results in the flow versus time equation for a single particle size interval:

$$Q = Q_o e^{-\alpha n^2 t} \qquad (3\text{-}9)$$

Solving for time and allowing for full particle size distribution to be reflected gives the following reference contaminant life equation:

$$T = -\ln (Q_F/Q_o) \Big/ \sum_{i=1}^{i_{max}} \alpha_i n_i^2 \qquad (3\text{-}10)$$

where Q_o and Q_F are respectively the initial and final flow rates and T is the contaminant service life. The α_i's are the reference contaminant wear coefficients evaluated from laboratory tests, and the n_i's are the particle concentrations for the various size intervals which describe the particle size distribution of the field fluid.

CONTAMINANT TOLERANCE PROFILE

The contaminant life equation can be utilized to calculate the pump contaminant life when the particle size distribution is known. In many instances, however, it is desirable to determine the maximum particle size distributions which would result in a specific contaminant life. Such information is required when determining the degree of filtration necessary to protect a given pump. In order to provide such information, the contaminant tolerance profile was developed.

The contaminant tolerance profile can be described as the locus of tangency points associated with particle size distribution lines which yield the same contaminant life. The profile is plotted on the conventional log-log² particle size distribution graph to facilitate comparison with various field distributions. In order to construct a profile, several different distributions must be found which yield the same contaminant life for the component. Finding these distributions is an iterative process which necessitates the use of a computer program [5]. The mechanics of the profile construction are illustrated in Fig. 3-2. The interpretation of the profile is simply that any straight-line distribution which is tangent to or below a contaminant tolerance profile will result in a contaminant life equal to or greater than the life associated with the profile itself.

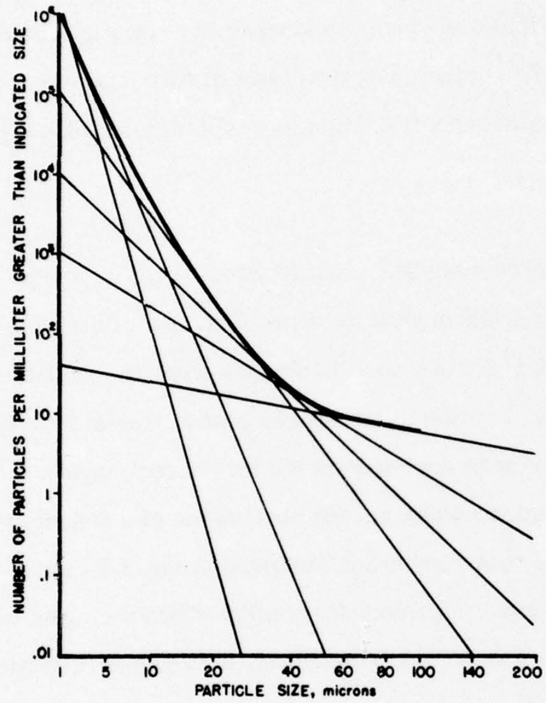

Fig. 3-2. Contaminant Profile Construction.

CHAPTER IV

CONTAMINANT LIFE TESTING

The basic procedure to be followed for determination of pump contaminant life for verification of the contaminant wear theory consisted of the following:

1. **Conduct contaminant sensitivity test on subject pump.**
2. **Calculate contaminant tolerance profile for pump.**
3. **Determine contamination level to result in desired life.**
4. **Select proper filter and ingression rate to produce required contamination level.**
5. **Conduct life test.**
6. **Compare results of life test to prediction.**

The first pump to be evaluated was a gear pump designated as FPRC Pump 223. A contaminant sensitivity test was conducted on this pump in accordance with the procedure described in Chapter II. The results of this test are included in Appendix A, and the one hundred hour contaminant tolerance profile calculated for this pump is shown in Fig. 4-1. A one hundred hour life was selected for the first few tests in order to provide a more rapid determination of the critical test parameters.

A filter was selected for use in the tests which had an extremely high contaminant capacity in order to reduce the number of element changes required. A standard multi-pass test [6] was conducted on this element to determine its filtration performance. The results of this test are presented in Fig. 4-2. The contamination level downstream of the filter at the first sample point is plotted in Fig. 4-1 with the 100-hour pump profile. Since the

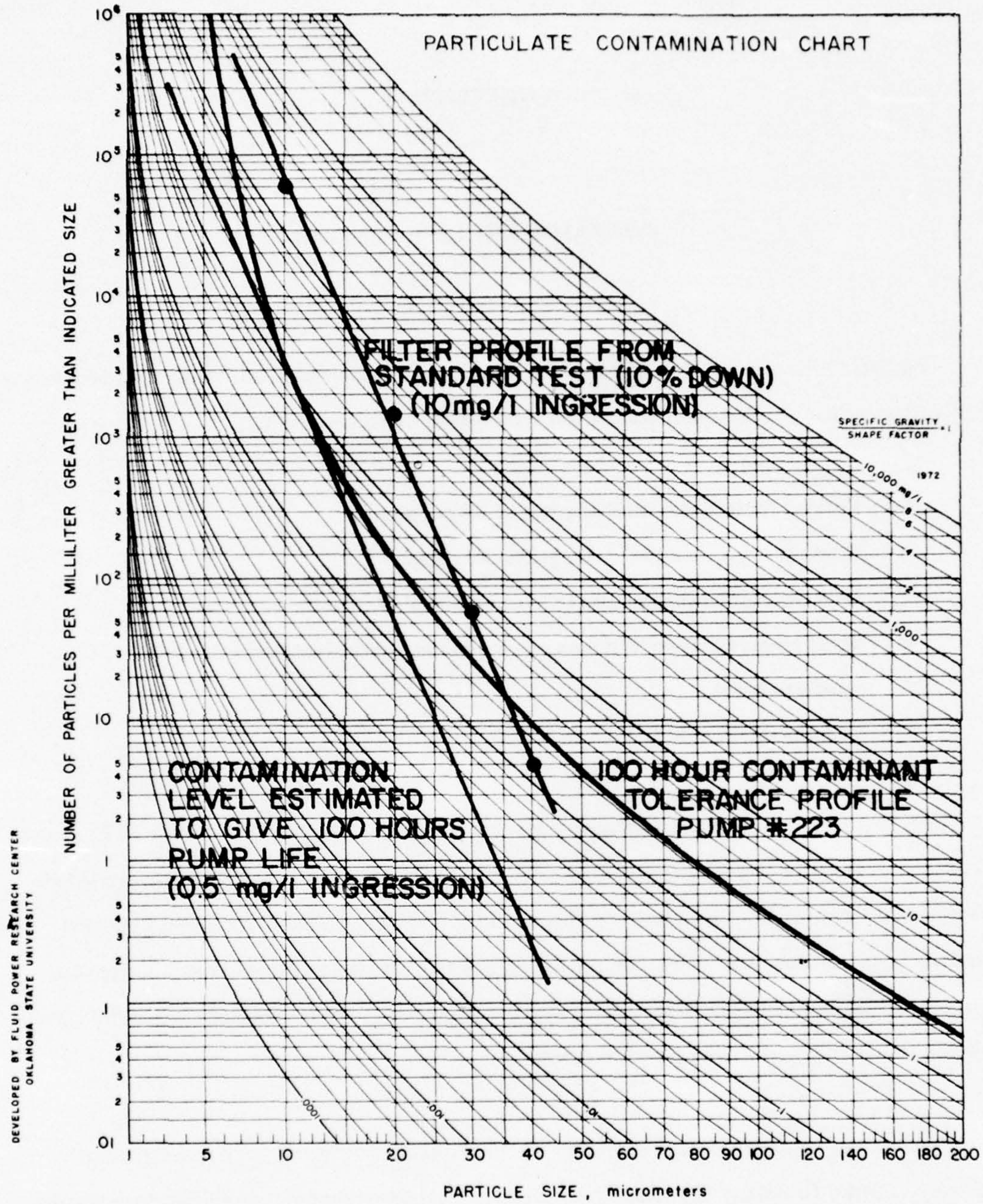

Fig. 4-1. Filter Protection Required for Pump No. 223.

FILTER: FPRC NUMBER 385B   DATE TESTED: 15 MARCH 1975
TEST LOCATION: FPRC - OSU   TEST FLOW RATE: 151.4 LPM

PRESSURE DROPS EXPRESSED IN UNITS OF BAR : TERMINAL 2.8, HOUSING 1.5, CLEAN ASSEMBLY 1.6, ELEMENT 0.0, NET 2.7.

% NET DROP	5	10	20	40	80	100
ASSEMBLY DROP	1.7	1.8	2.1	2.7	3.8	4.3
TIME (MIN.)	159.5	171.8	178.8	185.0	191.5	194.5

INJECTION FLUID	INITIAL	FINAL	AVERAGE
INJECTION FLOW RATE (LPM)	0.250	0.250	0.250
GRAVIMETRIC LEVEL (MG/L)	6337.6	6034.8	6186.2

FABRICATION INTEGRITY VERIFIED? YES, B.P. = 0.5 IN. WATER
INITIAL SYSTEM CLEANLINESS 3.6 PARTICLES/ML. > 10 MICRONS
BASE UPSTREAM GRAVIMETRIC LEVEL 10.21 MILLIGRAMS PER LITRE

PARTICLE DISTRIBUTION ANALYSIS
(AVG. NUMBER/ML. GREATER THAN INDICATED SIZE IN MICROMETRES)

SAMPLE		10	20	30	40	BETA 10
10%	UP	71858.0	1839.0	118.00	17.60	1.17
	DOWN	61584.0	1514.0	60.60	5.00	
20%	UP	69943.0	2621.0	172.00	23.40	1.09
	DOWN	63877.0	2396.0	117.60	5.60	
40%	UP	64745.0	3236.0	217.00	25.20	1.01
	DOWN	63878.0	3202.0	195.40	9.00	
80%	UP	37067.0	2077.0	166.40	26.00	1.02
	DOWN	36346.0	2064.0	144.40	8.80	

MINIMUM BETA 10 1.01   ACFTD CAPACITY 300.8 GM   FINAL LEVEL IN RESERVOIR 167.2 MG/L

Fig. 4-2. Filter Element Multi-Pass Test Report Sheet.

standard filter test was performed with a base upstream gravimetric level (ingression level) of ten milligrams/litre, it was estimated that an ingression of approximately 0.5 milligrams/litre would produce a contamination level just below the 100-hour profile for Pump No. 223. This is illustrated in Fig. 4-1.

A simplified schematic of the test circuit utilized for the pump contaminant life tests is given in Fig. 4-3. The main pump test system consists basically of a flowing loop with a load valve provided to produce the required test pressure. The contaminant conditioning system is similar to a multi-pass filter test system with continuous ingression provided by the contaminant injection system. For the pump life test on Pump No. 223, the flow rate in the contaminant conditioning system was 26 litres per minute, and the injection rate was set at approximately 13 milligrams per minute of AC Fine Test Dust. This resulted in the desired base upstream level at the filter of 0.5 milligrams/litre. The test configuration utilized was selected to produce longer filter life while retaining an approximation to a realistic field system. Mil-L-2104 cl. 10 hydraulic fluid at 65.5°C was utilized in all testing.

The actual contamination level obtained during the contaminant life test on the first Pump No. 223 varied somewhat throughout the test, as illustrated in Fig. 4-4. The actual particle counts are listed in Appendix B and are plotted in Fig. 4-5. It can be seen from Figs. 4-4 and 4-5 that the contamination level actually started out low and then built up to in excess of the level estimated from Fig. 4-1. The particle counts reported were obtained with an automatic particle counter calibrated with AC Fine Test Dust in accordance with the national standard [7].

The results of the actual contaminant life test conducted on Pump No. 223A are plotted in Fig. 4-6. The measured performance parameter was output flow with speed, pressure, and temperature maintained constant at 1800 RPM, 138 bar (2000 psi), and 65.5°C, respectively. The flow degradation ratio plotted in Fig. 4-6 is the ratio of the flow rate at the reference time to the initial or rated flow. A flow reduction of 20% (or a flow degradation ratio of 0.8) is

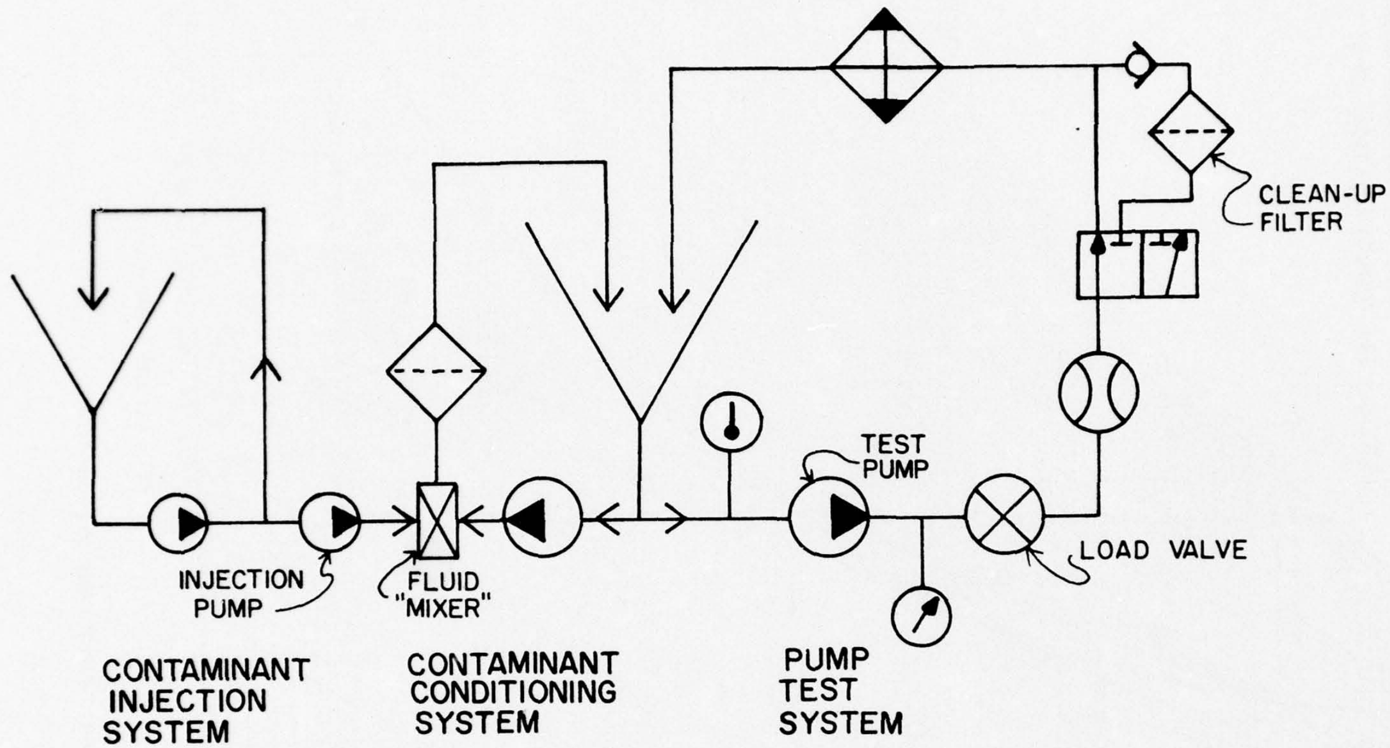

Fig. 4-3. Pump Contaminant Life Test Circuit.

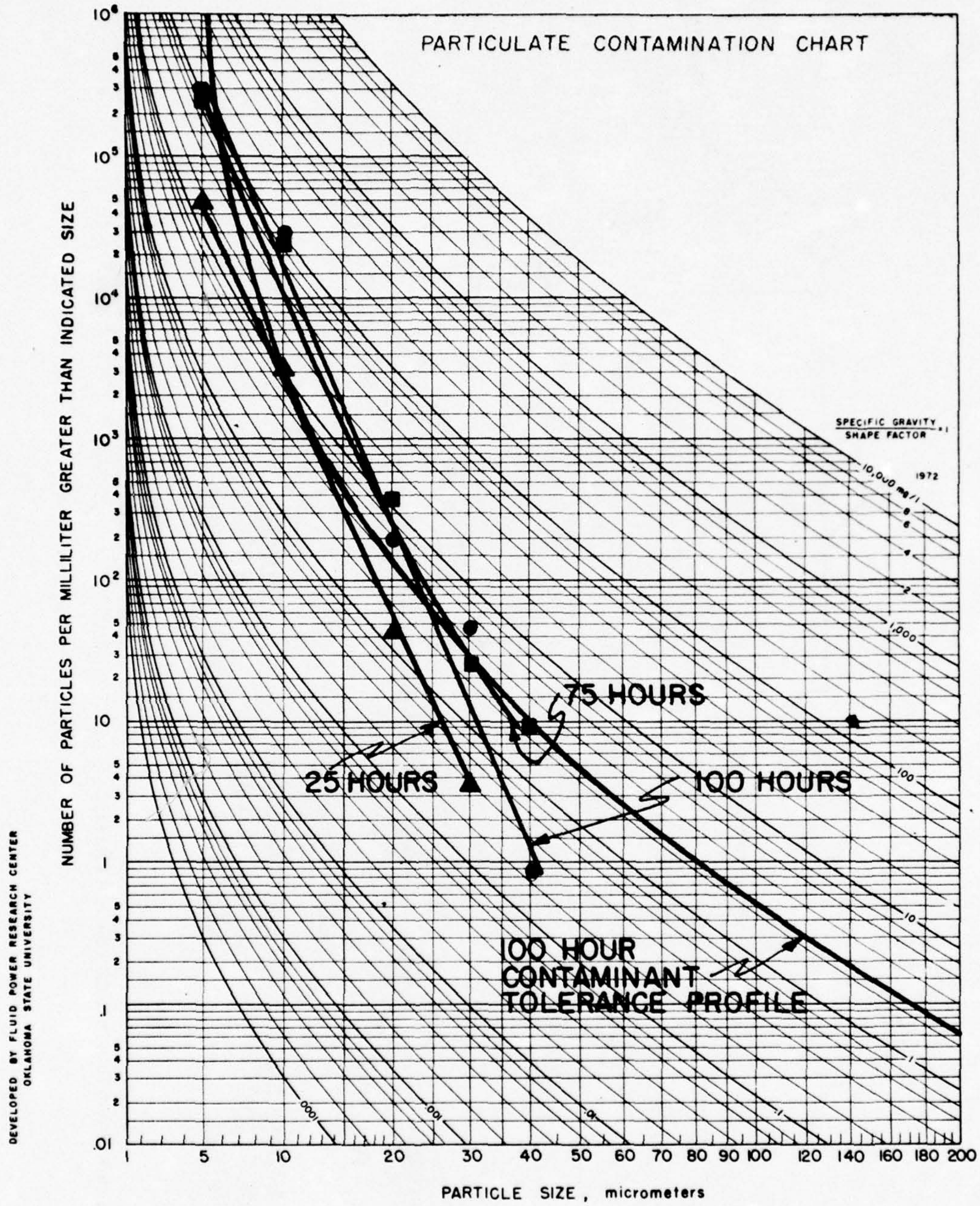

Fig. 4-4. Test Contamination Levels on Pump 223A.

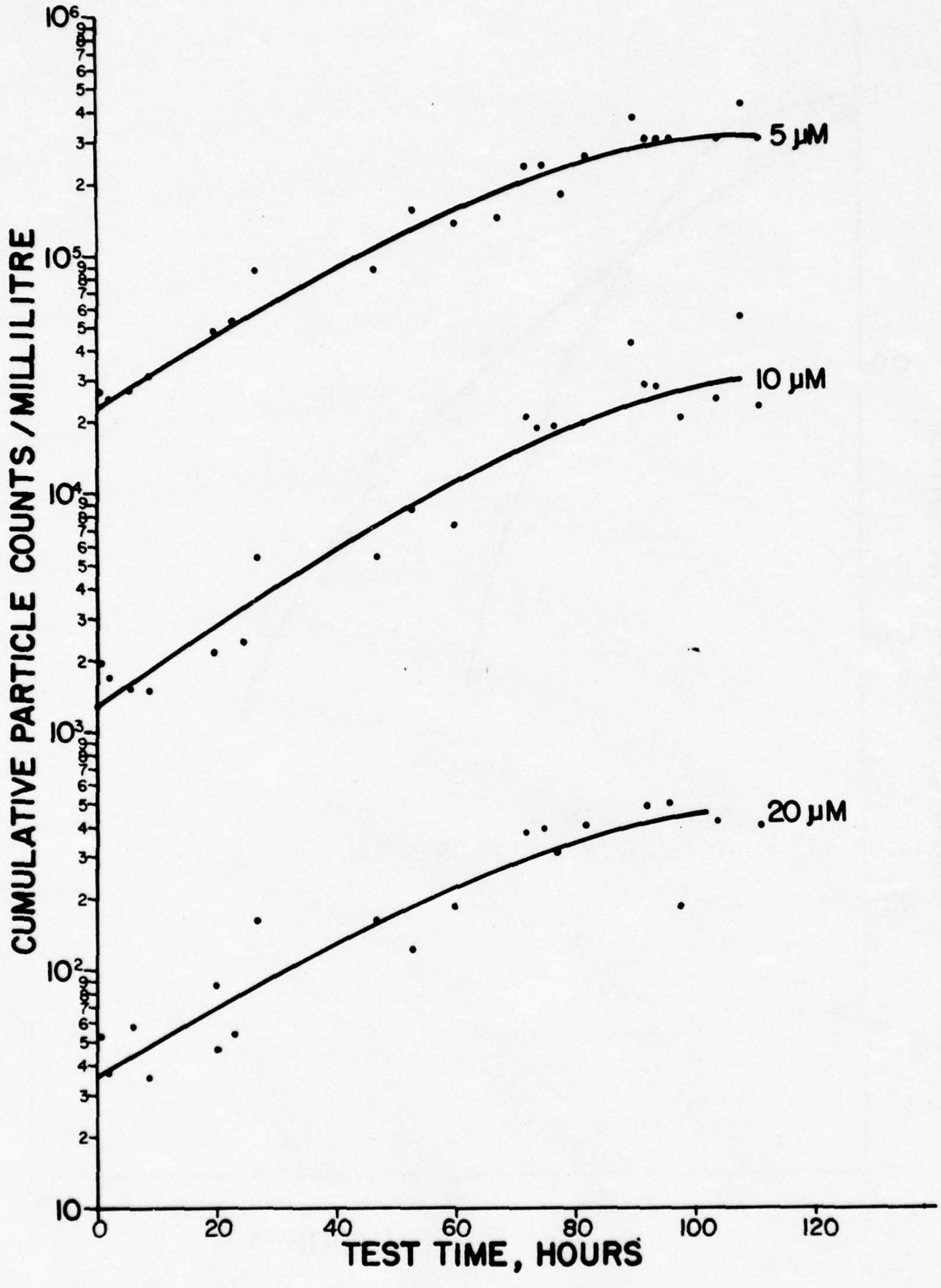

Fig. 4-5. Particle Count Levels Measured During Test on Pump No. 223A.

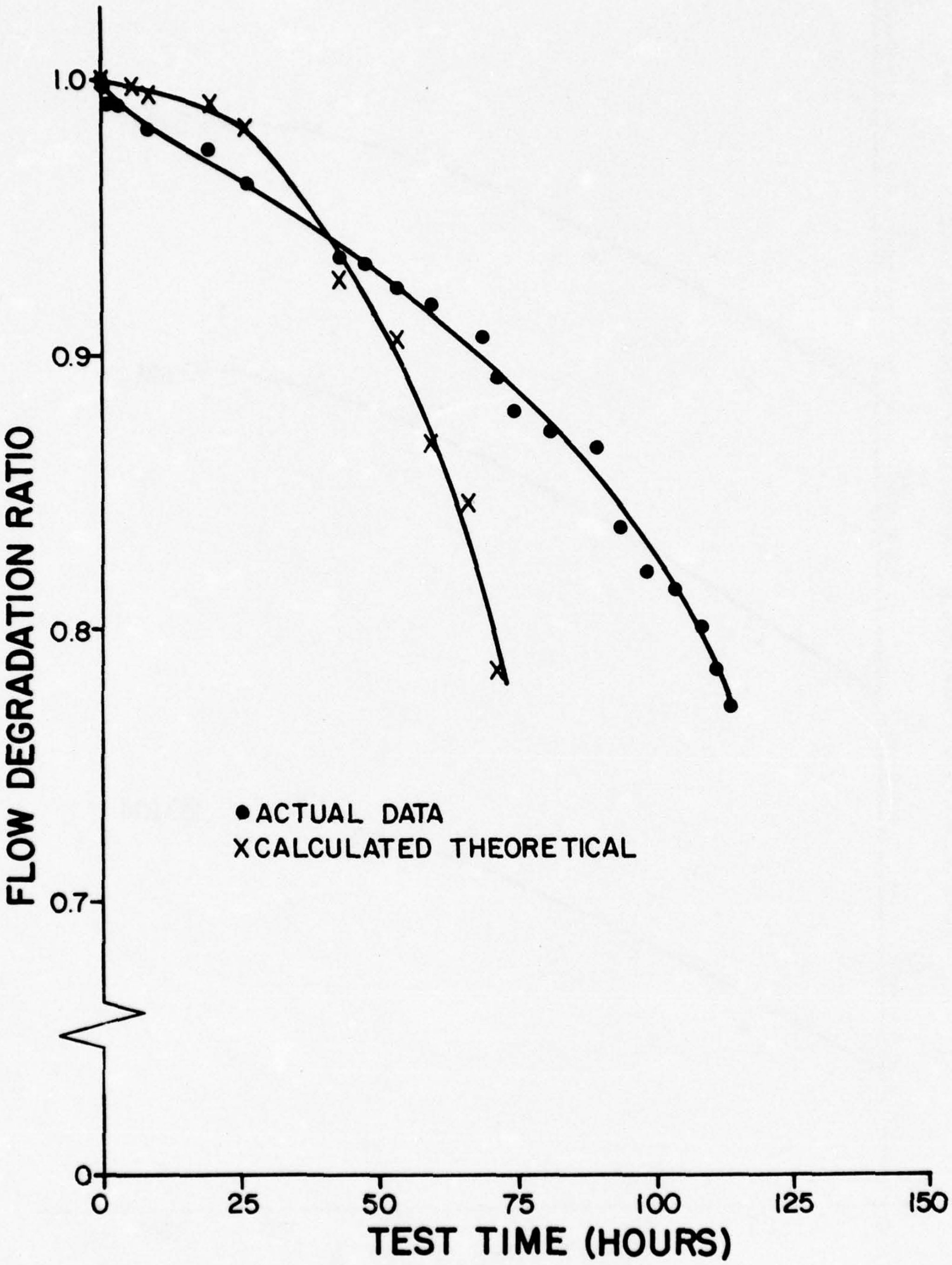

Fig. 4-6. Contaminant Life Results, Pump No. 223A.

generally considered failure; thus, Pump No. 223A reached failure after approximately 108 hours of operation.

Also plotted in Fig. 4-6 is the theoretical flow degradation curve. This curve was obtained by the use of Eq. (3-9) from Chapter III. Eq. (3-9) can be rewritten with summation for the various particle size intervals as follows:

$$Q \quad = \quad Q_o \, e^{-(\Sigma\alpha_i n_i^{\,2})t} \tag{4-1}$$

The values for the particle concentrations (n_i) at various times (t) can be found in Appendix B. Counts were utilized in the range between 1 and 200μM, and straight-line (log-log^2) extrapolation was utilized when necessary to obtain the required particle counts. The values for the contaminant wear coefficients (α) for the pump were obtained from a standard contaminant sensitivity test, as reported in Appendix A. Eq. (4-1) is utilized by setting Q_o equal to the initial or rated flow for the first time interval. A value for Q at the end of that time period is then calculated. Utilizing that new flow value, Q, and new particle concentrations, another flow degradation can be calculated for the next time interval. This process is continued for each time period where particle concentrations are available. The flow degradation ratios, as plotted in Fig. 4-6, are then calculated by dividing each of the final flow rates by the initial flow at the start of the test. The theoretical contaminant life for Pump No. 223A based upon the test conditions is approximately 72 hours, as seen from Fig. 4-6. This corresponds to a flow degradation of 20%.

Before initiating further contaminant life tests, the test system was revised to allow the simultaneous testing of up to three pumps. The test circuit was changed, as shown in Fig. 4-7. In this configuration, all test pumps are exposed to the identical contamination level, pressure, and temperature. A test utilizing this circuit was conducted on Pump No. 223B, No. 241, and No. 247. All three pumps were gear pumps and were operated at 138 bar, 1800 rpm, and 65.5°C with Mil-L-2104 hydraulic oil. Pump No. 223B was identical to Pump No. 223A. The

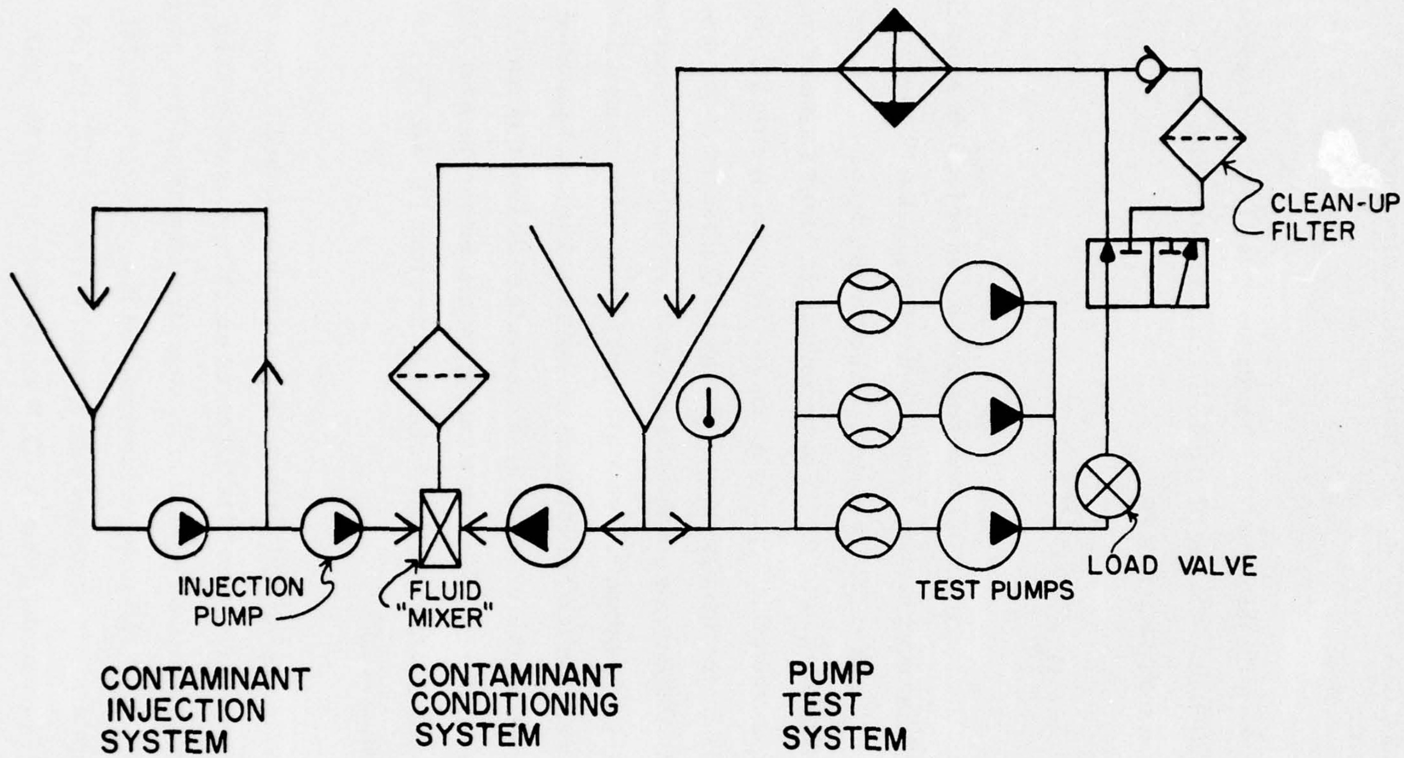

Fig. 4-7. Multiple Pump Life Test Circuit.

results of contaminant sensitivity tests on Pumps 241 and 247 are included in Appendix A. In addition, the actual particle count data obtained from the test are included in Appendix B.

The 100-hour contaminant tolerance profile for Pump No. 223B is shown in Fig. 4-8, along with some of the contamination levels measured throughout the test. Fig. 4-9 shows the actual flow degradation results which produced a contaminant life of approximately 122 hours. The calculated life based upon the test conditions was approximately five hours.

Figs. 4-10 and 4-11 illustrate similar data for Pump No. 241. The actual contaminant life and calculated *"theoretical"* life were approximately 262 hours and 53 hours, respectively. The test data for Pump No. 247 are shown in Figs. 4-12 and 4-13. For this pump, the actual life was 83 hours, while the calculated value was approximately one hour.

Based upon these test results, it appeared that running three pumps simultaneously with the same fluid (and contaminant) may have an unanticipated influence on the results. This was reasoned from the wide difference in calculated and actual contaminant life. The next test was therefore conducted as a single test on Gear Pump No. 252. The test conditions were similar to those utilized on the previous tests. The pump contaminant sensitivity data are included in Appendix A, and the particle counts are presented in Appendix B. Fig. 4-14 illustrates the 100-hour contaminant tolerance profile for Pump No. 252 as well as some of the test contamination levels. Fig. 4-15 shows the results of the actual test, which produced a contaminant life of approximately 123 hours. The calculated life was 57 hours based upon the test conditions.

Additional tests were attempted on two other pumps; however, these pumps failed before the test could be initiated. It is not known whether the failures were contamination related; however, additional tests on these pumps are being conducted. The results of standard contaminant sensitivity tests on these pumps (Nos. 242 and 243) are included in Appendix A for future reference.

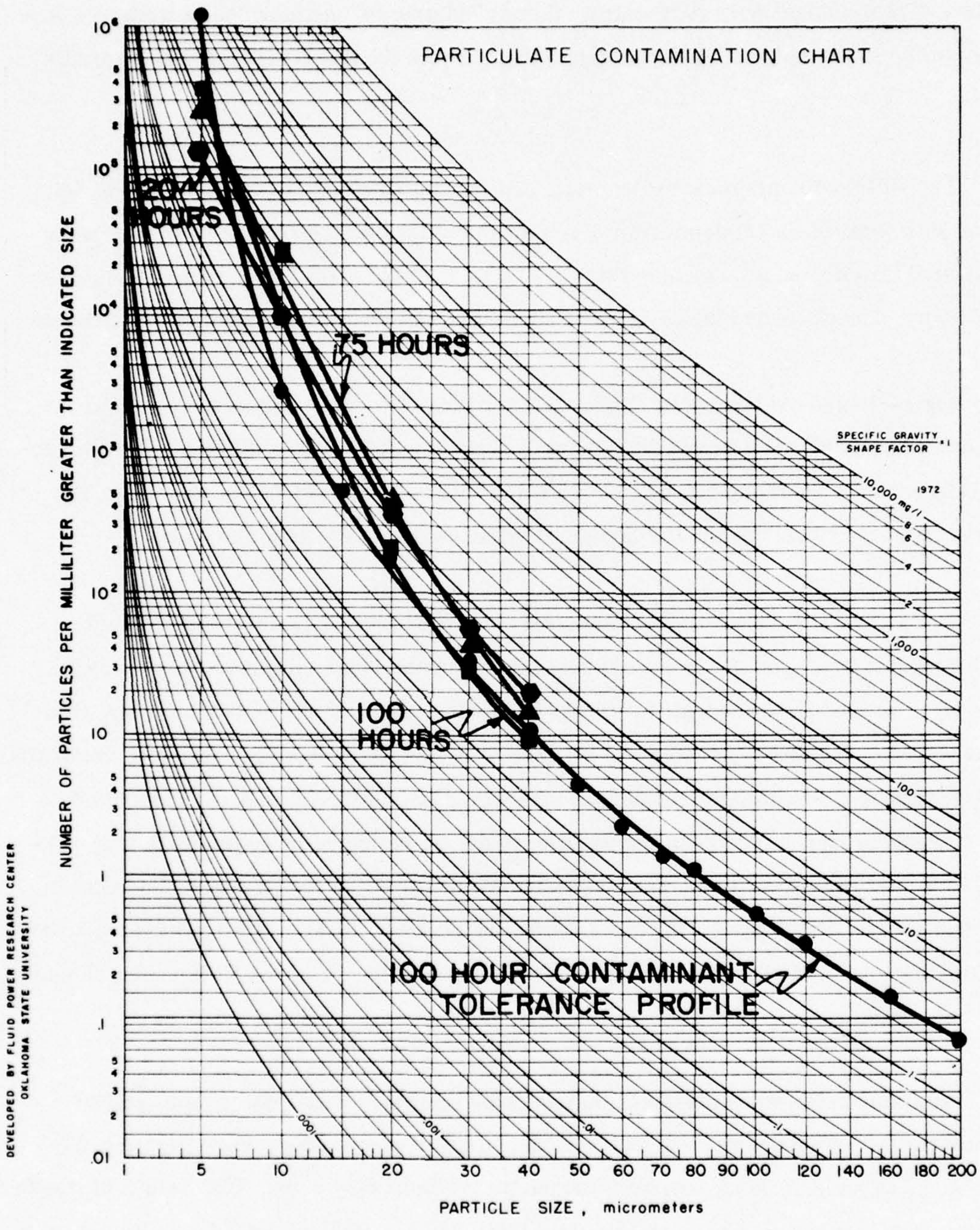

Fig. 4-8. Test Contamination Levels for Pump No. 223B.

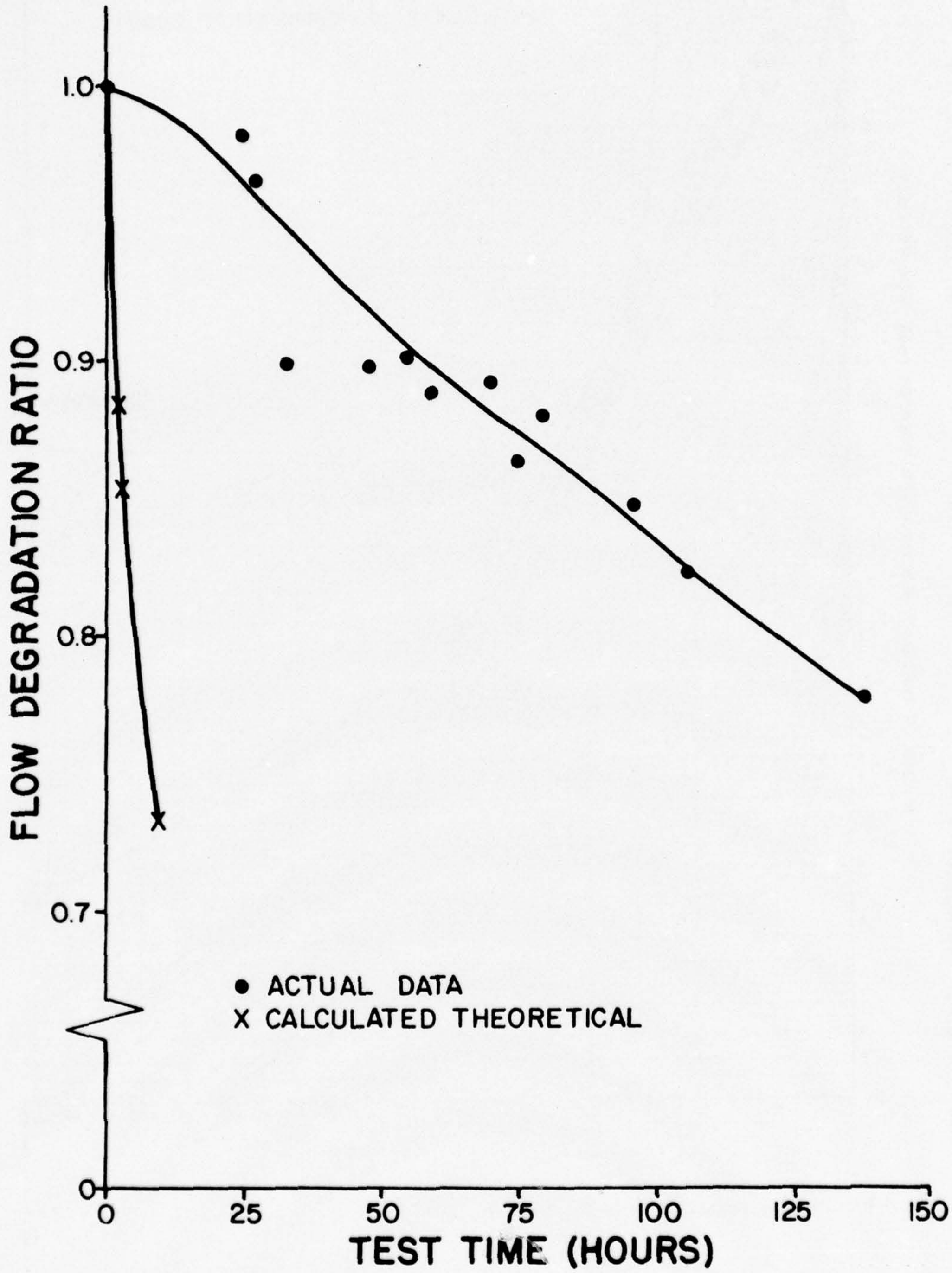

Fig. 4-9. Contaminant Life Results, Pump No. 223B.

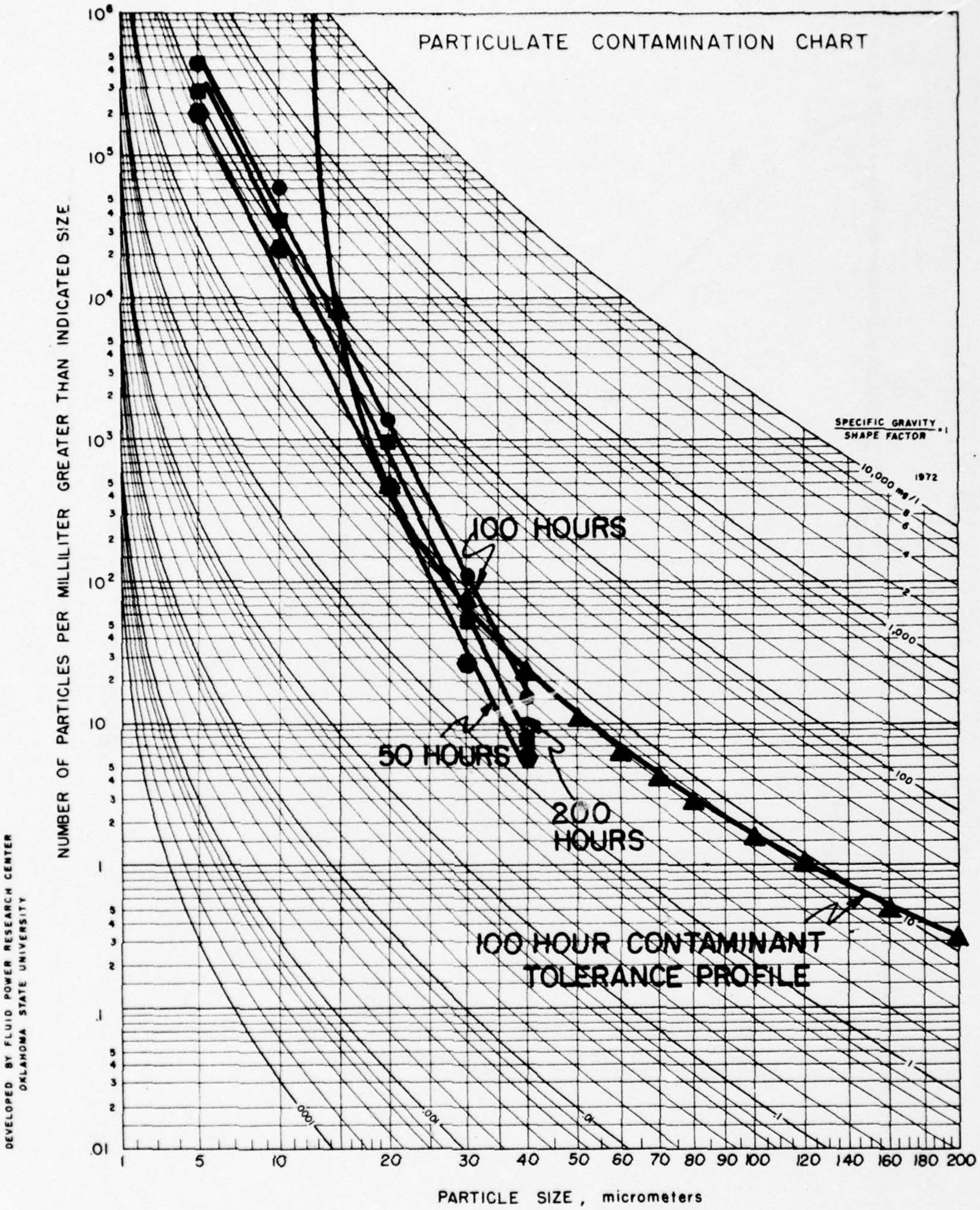

Fig. 4-10. Test Contamination Levels for Pump No. 241.

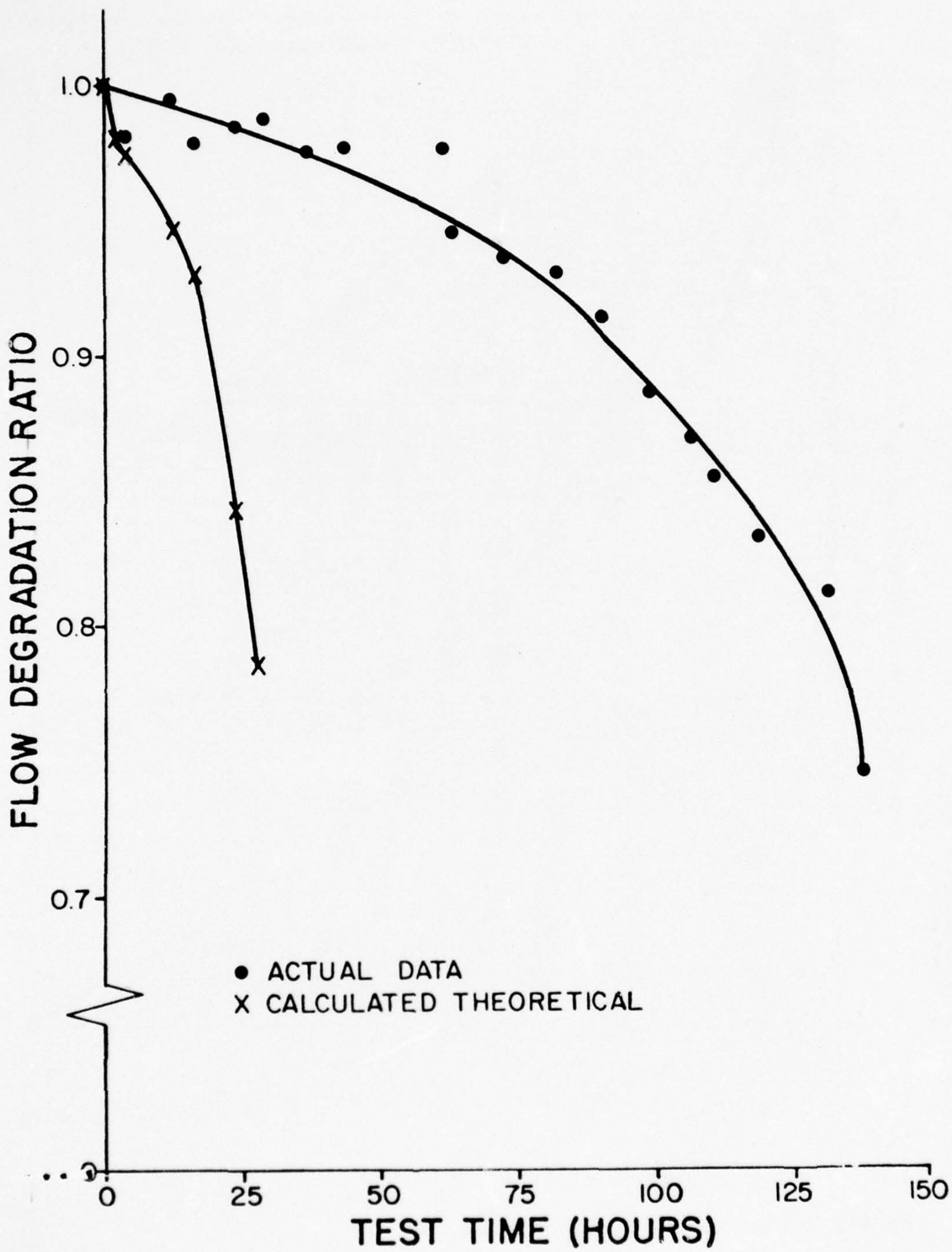

Fig. 4-11. Contaminant Life Results, Pump No. 241.

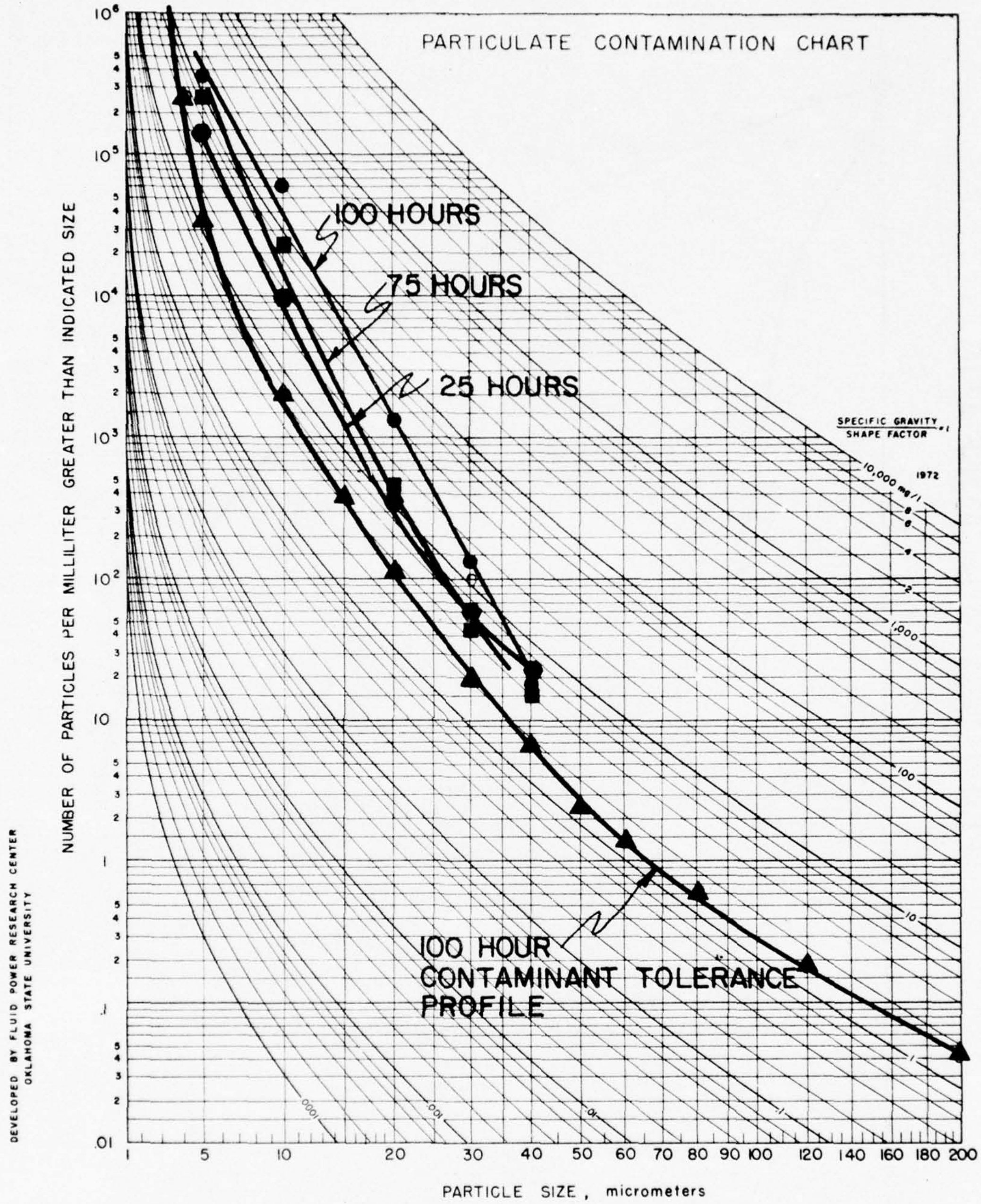

Fig. 4-12. Test Contamination Levels for Pump No. 247.

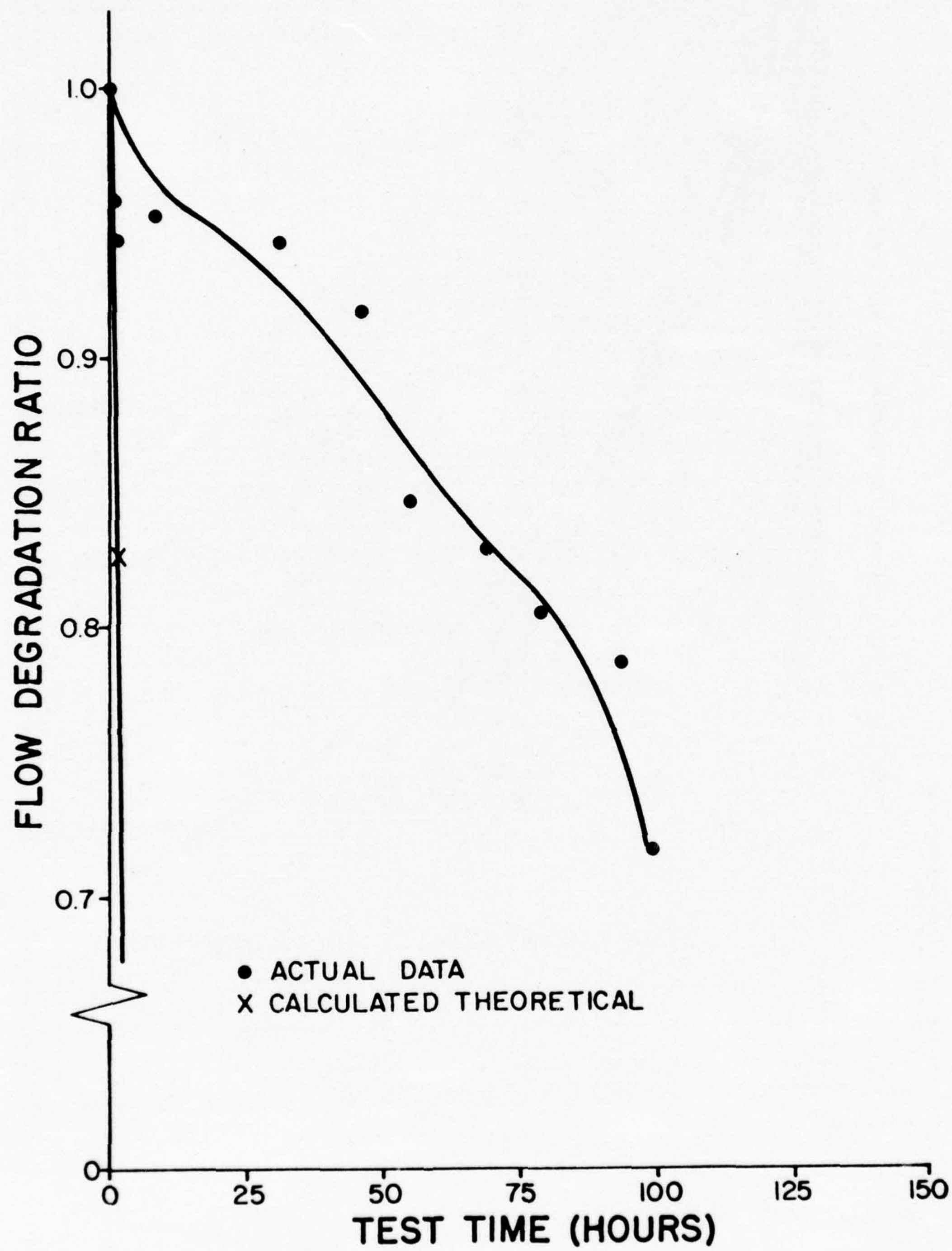

Fig. 4-13. Contaminant Life Results, Pump No. 247.

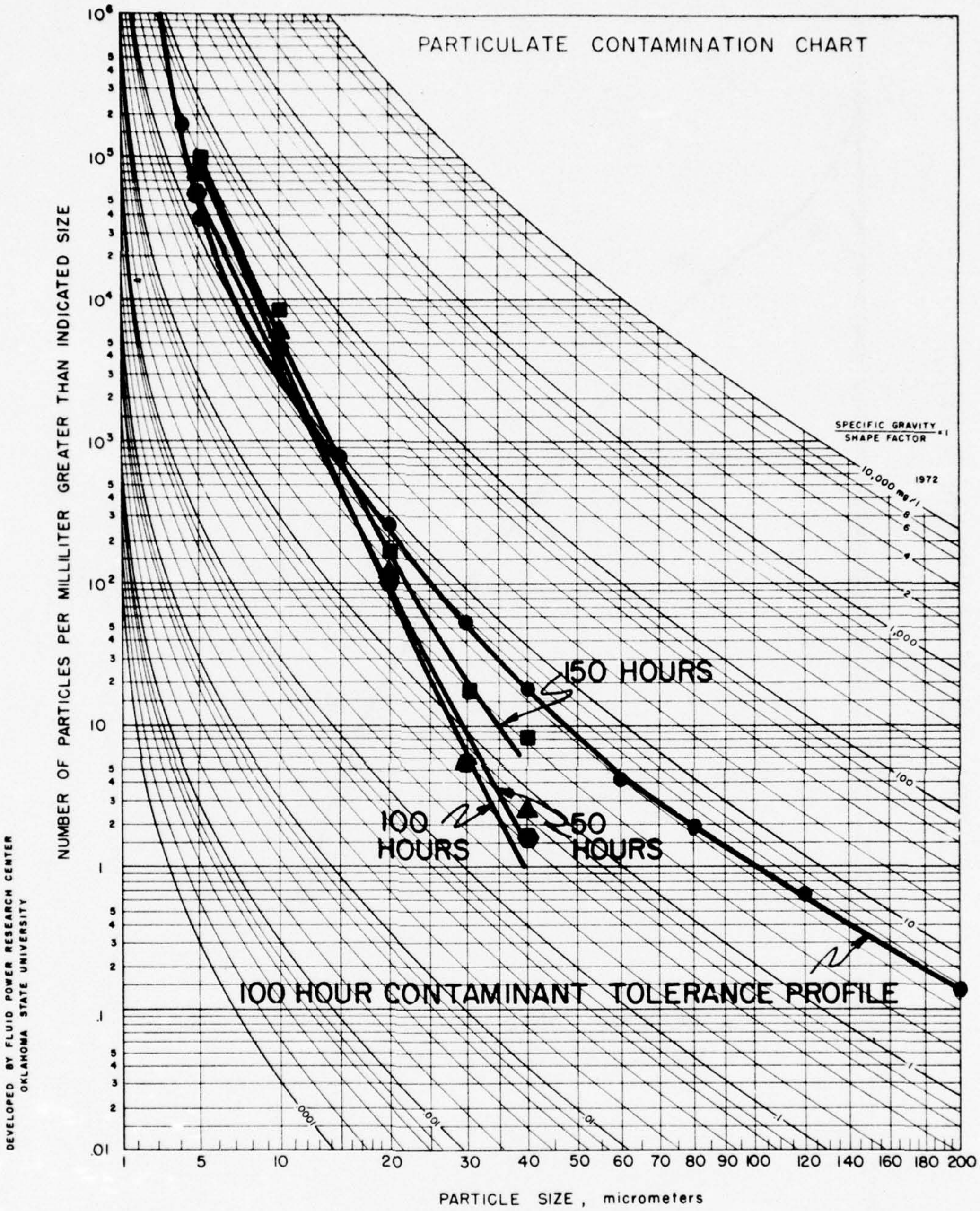

Fig. 4-14. Test Contamination Levels for Pump No. 252.

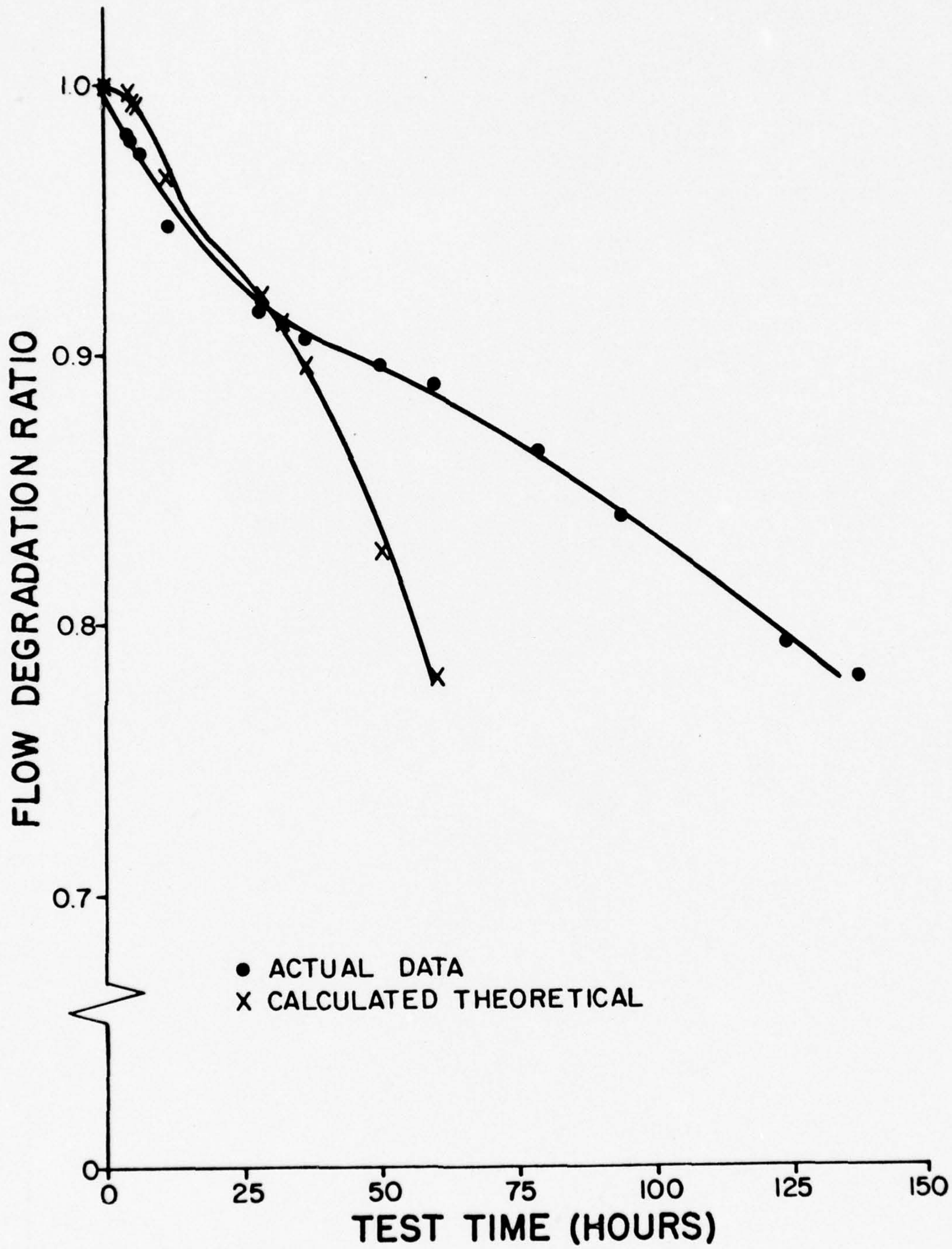

Fig. 4-15. Contaminant Life Results, Pump No. 252.

CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

The primary objective of this project was to verify or modify the pump contaminant wear theory previously developed at Oklahoma State University. With such a theory, filter protection levels could be correctly specified for MERDC systems to enhance their reliability and produce longer component life. The basic approach was to conduct long-term tests on hydraulic pumps with controlled operating parameters and contaminant environment. The pump life thus measured could then be compared to the life calculated using the contaminant wear relationships and pump sensitivity characteristics as measured by a standard accelerated test.

The first pump tested (No. 223A) exhibited a life of 108 hours until the flow had degraded by 20% of its original value. The contaminant life calculated using the wear relationships and actual particle counts was equal to approximately 72 hours. The calculated life was 67% of the actual life. It was felt that this was extremely close agreement when all the possible operating variables are considered.

After a relatively successful first test, it was decided to test three pumps simultaneously. The results on Pump Nos. 223B, 241, and 247 were not nearly as encouraging as Pump No. 223A. Pump No. 223B exhibited a calculated life (five hours) of only slightly more than 4% of the actual life obtained (122 hours). This pump was intentionally tested in order to determine the variations from test to test, especially when testing more than one pump at a time. When the 4% estimation is compared to the 67% obtained on an identical pump (No. 223A), a major difference in test results is revealed. The additional two pumps tested (Nos. 241 and 247) exhibited similar characteristics to No. 223B, with the calculated life representing

respectively 20% and 1.2% of the actual lives obtained. These values are much lower than the 67% obtained on Pump No. 223A tested as a single pump.

After a re-examination of the mathematical relationships presented in Chapter III, at least one possible explanation was found for the wide variation between calculated and actual life for the three pumps tested together. The laboratory contaminant wear equations assume that the contaminant is destroyed by the pump in an exponential manner with time constant τ. In reality, the contaminant is not actually destroyed, but its abrasive characteristics are probably altered (sharp edges smoothed, etc.) by the wear process. Thus, although the particles are still in existence, their effectiveness for causing wear has been lost. This effect has been recognized in the laboratory when wear stops for a given injection, but subsequent injections of the same size particles produce additional wear. The laboratory expressions are probably correct with the inclusion of a particle destruction time constant τ because the same wear reducing effect is actually present.

The field contaminant sensitivity expressions, however, do not include a particle destruction factor. In a real system or in the simulated test, there must be some *"particle destruction"* occurring, as assumed in the laboratory equations due to the multi-passing of the contaminated fluid. It is therefore not accurate to specify the contamination characteristics by only particle size distribution. Some account must be made of the effectiveness of the particles to cause wear. This can be accomplished by consideration of the filter and ingression characteristics in the field contaminant sensitivity expressions. In the test reported in which the three pumps were tested simultaneously, the rate at which the particles were *"destroyed"* must have been extremely high due to the fact that all three pumps were *"attacking"* the same contaminant. This could certainly explain the differences in the test results and calculated life.

In addition to the pump tests mentioned, one other test was completed. This pump (No. 252) was tested individually and exhibited agreement between calculated and actual life more nearly like the first pump tested. The theoretical life of 57 hours was 46% of the actual life of 123 hours. Further refinement of the contaminant sensitivity expressions should produce even closer agreement.

Several additional tests are planned for the continuation of this project. These include tests on the pumps which exhibited premature failure and long-term (500-1000 hour) tests on all the pumps. The contaminant sensitivity expressions will be modified to include a field particle destruction rate as the required data are collected to define the process.

The primary difficulty encountered when conducting the pump life tests was maintaining the test facility in an operational condition. The severe environment was a test upon the other system components as much as the pump under evaluation. It is felt that the test facility is now reliable and suitable for conducting long-term tests.

Overall, this verification study has been successful. Although additional data are necessary to accurately refine the contaminant sensitivity theory, the results of the project can certainly be utilized to indicate trends. In all instances, the actual pump life obtained was in excess of the calculated value. Because of this conservative estimation, one can certainly feel safe in utilizing the theory and equations as they exist. However, future testing should provide the necessary refinement to allow closer estimation of actual life.

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APPENDIX A

PUMP CONTAMINANT SENSITIVITY TEST DATA

FPRC PUMP NO. 223, 1800 RPM, 2000 PSI Q RATED= 19.17 GPM

DIAMETER	QF	QF/QR	QF/QO	ALPHA
5.0	19.17	1.000	1.000	0.0
10.0	19.17	1.000	1.000	0.0
20.0	18.05	0.942	0.942	5.6477E-12
30.0	16.03	0.836	0.888	4.0973E-10
40.0	13.79	0.719	0.860	4.4665E-09
50.0	10.34	0.539	0.750	1.3045E-07
60.0			0.660	7.6811E-07
70.0			0.562	4.5721E-06
80.0			0.465	2.1593E-05
90.0			0.372	8.6258E-05
100.0			0.288	3.0130E-04

INTERCEPT = 5.000E 04 GRAVIMETRIC = 3.200E 00 LIFE = 6.303E 02 HOURS

INTERCEPT = 3.200E 04 GRAVIMETRIC = 3.200E 00 LIFE = 4.329E 02 HOURS

INTERCEPT = 5.000E 03 GRAVIMETRIC = 3.200E 00 LIFE = 6.164E 01 HOURS

CONTAMINANT TOLERANCE PROFILE FOR LIFE OF 100.000 HOURS

SIZE	NO. > SIZE
200.00	7.1687E-02
160.00	1.4269E-01
120.00	3.3801E-01
100.00	5.6457E-01
80.00	1.0333E 00
70.00	1.4396E 00
60.00	2.2208E 00
50.00	4.3146E 00
40.00	1.0471E 01
30.00	3.3596E 01
20.00	1.6985E 02
15.00	5.1360E 02
10.00	2.5823E 03
5.00	1.2278E 06
4.00	1.6435E 08
3.00	1.0116E 11

FPRC PUMP NO. 241, 1800 RPM, 2000 PSI   Q RATED= 19.50 GPM

DIAMETER	QF	QF/QR	QF/QO	ALPHA
5.0	19.50	1.000	1.000	0.0
10.0	19.50	1.000	1.000	0.0
20.0	19.50	1.000	1.000	0.0
30.0	18.61	0.954	0.954	2.5781E-10
40.0	17.71	0.908	0.952	8.3798E-10
50.0	16.59	0.851	0.937	1.6999E-08
60.0	14.91	0.765	0.899	2.4471E-07
70.0	13.79	0.707	0.925	-7.0762E-07
80.0	12.55	0.644	0.910	1.8886E-06
90.0			0.830	3.5000E-05
100.0			0.818	1.7980E-05

INTERCEPT = 5.000E 04 GRAVIMETRIC = 3.200E 00 LIFE = 4.991E 03 HOURS

INTERCEPT = 3.200E 04 GRAVIMETRIC = 3.200E 00 LIFE = 2.422E 03 HOURS

INTERCEPT = 5.000E 03 GRAVIMETRIC = 3.200E 00 LIFE = 4.269E 02 HOURS

CONTAMINANT TOLERANCE PROFILE FOR LIFE OF 100.000 HOURS

SIZE	NO. > SIZE
200.00	2.0575E-01
160.00	4.1278E-01
120.00	9.5966E-01
100.00	1.6159E 00
80.00	2.9284E 00
70.00	4.1784E 00
60.00	6.1159E 00
50.00	1.1053E 01
40.00	2.5116E 01
30.00	7.6991E 01
20.00	4.1387E 02
15.00	7.8889E 03
10.00	1.2987E 07
5.00	1.7049E 13

FPRC PUMP NO. 242, 1800 RPM, 2000 PSI   Q RATED= 17.49 GPM

DIAMETER	QF	QF/QR	QF/QO	ALPHA
5.0	17.49	1.000	1.000	0.0
10.0	17.49	1.000	1.000	0.0
20.0	17.37	0.993	0.993	5.4587E-13
30.0	17.15	0.981	0.987	4.2301E-11
40.0	16.81	0.961	0.980	8.4762E-10
50.0	16.25	0.929	0.967	1.3470E-08
60.0	15.47	0.885	0.952	9.2609E-08
70.0	14.68	0.839	0.949	1.1677E-07
80.0	13.55	0.775	0.924	3.0131E-06
90.0			0.902	9.4260E-06
100.0			0.881	2.6958E-05

INTERCEPT = 5.000E 04 GRAVIMETRIC = 3.200E 00 LIFE = 5.537E 03 HOURS

INTERCEPT = 3.200E 04 GRAVIMETRIC = 3.200E 00 LIFE = 3.345E 03 HOURS

INTERCEPT = 5.000E 03 GRAVIMETRIC = 3.200E 00 LIFE = 6.198E 02 HOURS

CONTAMINANT TOLERANCE PROFILE FOR LIFE OF 100.000 HOURS

SIZE	NO. > SIZE
200.00	2.3309E-01
160.00	4.6747E-01
120.00	1.0925E 00
100.00	1.8268E 00
80.00	3.3268E 00
70.00	4.7013E 00
60.00	7.0768E 00
50.00	1.3577E 01
40.00	3.2327E 01
30.00	1.0233E 02
20.00	5.0358E 02
15.00	1.5255E 03
10.00	7.6504E 03
5.00	2.4242E 06
4.00	2.9480E 08
3.00	1.6422E 11

FPRC PUMP NO. 243, 1800 RPM, 2000 PSI Q RATED= 13.67 GPM

DIAMETER	QF	QF/QR	QF/QO	ALPHA
5.0	13.67	1.000	1.000	0.0
10.0	13.67	1.000	1.000	0.0
20.0	13.67	1.000	1.000	0.0
30.0	13.11	0.959	0.959	2.3082E-10
40.0	11.10	0.812	0.847	1.2363E-08
50.0	8.74	0.639	0.787	7.4337E-08
60.0			0.685	8.2367E-07
70.0			0.584	4.5138E-06
80.0			0.484	2.1291E-05
90.0			0.388	8.5131E-05
100.0			0.302	2.9708E-04

INTERCEPT = 5.000E 04 GRAVIMETRIC = 3.200E 00 LIFE = 2.590E 03 HOURS

INTERCEPT = 3.200E 04 GRAVIMETRIC = 3.200E 00 LIFE = 8.773E 02 HOURS

INTERCEPT = 5.000E 03 GRAVIMETRIC = 3.200E 00 LIFE = 6.327E 01 HOURS

CONTAMINANT TOLERANCE PROFILE FOR LIFE OF 100.000 HOURS

SIZE	NO. > SIZE
120.00	3.3856E-01
100.00	5.6512E-01
80.00	1.0339E 00
70.00	1.4714E 00
60.00	2.2214E 00
50.00	4.4401E 00
40.00	1.1534E 01
30.00	4.3096E 01
20.00	4.0622E 02
15.00	2.4287E 04
10.00	1.5173E 08
5.00	1.1658E 15

FPRC PUMP NO. 247, 1800 RPM, 2000 PSI Q RATED= 21.63 GPM

DIAMETER	QF	QF/QR	QF/QO	ALPHA
5.0	21.63	1.000	1.000	0.0
10.0	21.52	0.995	0.995	2.9421E-14
20.0	20.06	0.927	0.932	6.2995E-12
30.0	15.36	0.710	0.766	1.1867E-09
40.0	10.02	0.466	0.657	1.7788E-08
50.0			0.470	3.2012E-07
60.0			0.324	2.2197E-06
70.0			0.203	1.3181E-05
80.0			0.115	6.3506E-05
90.0			0.058	2.6176E-04
100.0			0.025	9.5295E-04

INTERCEPT = 5.000E 04 GRAVIMETRIC = 3.200E 00 LIFE = 3.460E 02 HOURS

INTERCEPT = 3.200E 04 GRAVIMETRIC = 3.200E 00 LIFE = 2.014E 02 HOURS

INTERCEPT = 5.000E 03 GRAVIMETRIC = 3.200E 00 LIFE = 2.067E 01 HOURS

CONTAMINANT TOLERANCE PROFILE FOR LIFE OF 100.000 HOURS

SIZE	NO. > SIZE
200.00	4.1375E-02
120.00	1.9317E-01
100.00	3.2208E-01
80.00	5.8770E-01
70.00	8.2989E-01
60.00	1.2674E 00
50.00	2.5330E 00
40.00	6.3455E 00
30.00	2.1283E 01
20.00	1.1535E 02
15.00	3.7160E 02
10.00	1.8903E 03
5.00	3.5640E 04
4.00	2.4064E 05
3.00	7.4156E 06
2.00	9.9892E 08
1.00	1.3620E 11

FPFC PUMP NO. 252, 1800 RPM, 2000 PSI   Q RATED= 22.75 GPM

DIAMETER	QF	QF/QR	QF/QD	ALPHA
5.0	22.75	1.000	1.000	0.0
10.0	22.64	0.995	0.995	2.7976E-14
20.0	22.31	0.981	0.985	1.1000E-12
30.0	20.96	0.921	0.939	2.8466E-10
40.0	18.72	0.823	0.893	5.5711E-09
50.0	17.60	0.774	0.940	-3.8110E-08
60.0	15.58	0.685	0.885	3.4775E-07
70.0			0.825	1.9908E-06
80.0			0.789	5.0607E-06
90.0			0.741	2.4487E-05
100.0			0.691	8.4112E-05

INTERCEPT = 5.000E 04 GRAVIMETRIC = 3.200E 00 LIFE = 1.294E 03 HOURS

INTERCEPT = 3.200E 04 GRAVIMETRIC = 3.200E 00 LIFE = 1.039E 03 HOURS

INTERCEPT = 5.000E 03 GRAVIMETRIC = 3.200E 00 LIFE = 2.116E 02 HOURS

CONTAMINANT TOLERANCE PROFILE FOR LIFE OF 100.000 HOURS

SIZE	NO. > SIZE
200.00	1.3544E-01
160.00	2.7216E-01
120.00	6.3934E-01
100.00	1.0612E 00
80.00	1.9362E 00
70.00	2.7487E 00
60.00	4.1237E 00
50.00	7.8737E 00
40.00	1.8436E 01
30.00	5.6561E 01
20.00	2.6656E 02
15.00	7.7906E 02
10.00	3.3353E 03
5.00	3.6960E 04
4.00	1.7790E 05
3.00	3.8029E 06
2.00	3.5868E 08
1.00	3.8659E 10

APPENDIX B

PARTICLE COUNT DATA FROM PUMP LIFE TESTS

TABLE B-1. CONTAMINATION LEVEL DATA FROM TEST ON PUMP NO. 223A.

PARTICLES PER ML. GREATER THAN INTENDED SIZE

TIME (hours)	> 5 MIC.	> 10 MIC.	> 20 MIC.	> 30 MIC.	> 40 MIC.
0.5	2.761E 04	4.965E 03	1.480E 02	1.300E 01	4.000E 00
1.0	2.560E 04	2.740E 03	5.350E 01	2.500E 00	5.000E-01
2.2	2.481E 04	2.417E 03	3.700E 01	2.500E 00	1.500E 00
3.2	2.557E 04	1.919E 03	1.140E 02	3.150E 01	1.500E 01
5.8	2.716E 04	2.107E 03	5.850E 01	1.600E 01	6.500E 00
8.7	3.126E 04	2.112E 03	3.600E 01	3.500E 00	5.000E-01
19.8	4.889E 04	3.172E 03	8.750E 01	2.250E 01	6.000E 00
20.8	4.937E 04	3.145E 03	4.700E 01	3.500E 00	1.000E-06
23.0	5.300E 04	3.388E 03	5.400E 01	9.500E 00	4.000E 00
27.1	8.833E 04	7.688E 03	1.610E 02	1.150E 01	3.500E 00
47.9	1.592E 05	1.188E 04	1.140E 02	3.000E 00	3.000E 00
53.3	1.563E 05	1.214E 04	1.220E 02	8.000E 00	2.500E 00
60.1	1.360E 05	1.018E 04	1.185E 02	9.500E 00	2.500E 00
66.9	1.441E 05	6.944E 03	4.300E 01	9.000E 00	4.000E 00
69.0	2.356E 05	2.826E 04	5.375E 02	5.400E 01	1.850E 01
70.7	2.393E 05	2.974E 04	4.785E 02	3.200E 01	1.150E 01
70.9	2.341E 05	2.638E 04	4.565E 02	4.850E 01	2.100E 01
72.0	2.390E 05	2.865E 04	3.800E 02	2.400E 01	7.500E 00
73.4	2.419E 05	2.632E 04	4.100E 02	2.750E 01	8.500E 00
75.0	2.416E 05	2.558E 04	3.875E 02	2.550E 01	9.500E 00
77.4	1.843E 05	2.635E 04	3.225E 02	2.300E 01	8.500E 00
81.7	2.549E 05	2.842E 04	4.075E 02	1.600E 01	3.500E 00
89.9	3.781E 05	6.058E 04	2.005E 03	8.720E 01	1.440E 01
91.9	3.178E 05	3.883E 04	4.872E 02	1.600E 00	8.000E-01
93.9	3.235E 05	3.743E 04	4.248E 02	1.440E 01	8.000E-01
95.9	3.131E 05	3.556E 04	4.992E 02	8.000E 00	3.200E 00
97.9	2.893E 05	2.899E 04	1.856E 02	4.800E 00	8.000E-01
103.9	3.156E 05	3.480E 04	4.248E 02	1.920E 01	2.400E 00
107.9	4.229E 05	7.865E 04	2.838E 03	9.920E 01	8.000E 00
111.2	2.993E 05	3.205E 04	4.050E 02	6.100E 01	2.400E 01
112.3	3.146E 05	3.266E 04	3.470E 02	3.300E 01	1.100E 01
114.5	2.829E 05	2.555E 04	1.910E 02	1.600E 01	7.000E 00
117.0	2.956E 05	3.058E 04	3.860E 02	3.000E 00	1.000E-06
119.5	2.905E 05	2.985E 04	4.230E 02	2.100E 01	8.000E 00
121.4	2.932E 05	2.951E 04	3.870E 02	1.200E 01	1.000E 00
129.2	3.874E 05	4.299E 04	9.000E 02	5.100E 01	1.100E 01
132.5	3.379E 05	4.057E 04	5.670E 02	1.900E 01	3.000E 00
134.8	3.273E 05	3.952E 04	6.210E 02	1.300E 01	1.000E 00
136.1	3.601E 05	4.843E 04	9.920E 02	2.300E 01	1.000E 00

TABLE B-2. CONTAMINATION LEVEL DATA FROM TEST ON PUMP NOS. 223B, 241, AND 247.

PARTICLES PER ML GREATER THAN INTENDED SIZE

TIME (hours)	> 5 MIC.	> 10 MIC.	> 20 MIC.	> 30 MIC.	> 40 MIC.
0.0	2.531E 05	4.066E 04	1.658E 03	1.040E 02	1.440E 01
1.3	1.898E 05	2.164E 04	3.992E 02	3.120E 01	6.400E 00
2.2	1.885E 05	1.673E 04	1.968E 02	9.600E 00	1.600E 00
8.0	1.892E 05	1.709E 04	3.200E 02	2.240E 01	8.800E 00
23.1	1.341E 05	9.236E 03	3.630E 02	5.900E 01	2.000E 01
32.4	1.944E 05	2.403E 04	3.700E 02	1.400E 01	1.000E 00
46.5	3.135E 05	4.116E 04	1.044E 03	4.700E 01	7.000E 00
54.7	1.984E 05	2.084E 04	4.830E 02	2.900E 01	6.000E 00
58.2	2.488E 05	1.886E 04	3.350E 02	4.800E 01	1.400E 01
69.4	2.376E 05	2.265E 04	4.710E 02	3.500E 01	5.000E 00
78.0	2.753E 05	2.377E 04	4.330E 02	4.200E 01	1.500E 01
93.9	2.877E 05	2.737E 04	5.110E 02	4.600E 01	7.000E 00
98.1	4.492E 05	6.171E 04	1.423E 03	1.030E 02	1.600E 01
105.4	3.622E 05	2.676E 04	2.040E 02	3.000E 01	9.000E 00
120.6	4.257E 05	4.787E 04	1.042E 03	5.600E 01	1.000E 01
122.2	7.820E 05	1.226E 05	3.198E 03	1.100E 02	1.000E 01
126.2	1.887E 05	3.499E 04	9.572E 02	3.080E 01	3.200E 00
143.0	1.152E 05	1.310E 04	2.316E 02	1.440E 01	2.400E 00
149.4	1.174E 05	1.190E 04	3.904E 02	4.160E 01	1.280E 01
164.3	1.918E 05	2.266E 04	6.152E 02	3.040E 01	7.200E 00
175.2	2.332E 05	2.817E 04	7.688E 02	5.760E 01	1.360E 01
188.4	2.426E 05	2.540E 04	5.120E 02	1.600E 01	3.200E 00
197.0	2.779E 05	3.279E 04	9.760E 02	5.200E 01	7.200E 00
212.0	3.246E 05	4.117E 04	7.368E 02	2.720E 01	6.400E 00
218.3	4.278E 05	6.497E 04	1.738E 03	9.600E 01	6.400E 00
236.7	3.797E 05	5.420E 04	1.113E 03	2.880E 01	2.400E 00
261.9	2.621E 05	2.526E 04	4.440E 02	4.400E 01	9.600E 00
266.0	1.025E 06	2.015E 05	1.170E 04	7.960E 02	5.800E 01

TABLE B-3. CONTAMINATION LEVEL DATA FROM TEST ON PUMP NO. 252.

PARTICLES PER ML GREATER THAN INTENDED SIZE

TIME (hours)	> 5 MIC.	> 10 MIC.	> 20 MIC.	> 30 MIC.	> 40 MIC.
0.1	1.149E 04	2.527E 03	1.663E 02	2.630E 01	8.330E 00
3.5	2.125E 04	2.438E 03	1.037E 02	1.300E 01	5.330E 00
4.7	2.613E 04	3.010E 03	1.480E 02	1.700E 01	5.330E 00
5.9	3.503E 04	4.504E 03	1.980E 02	2.170E 01	5.330E 00
6.3	3.987E 04	5.572E 03	3.450E 02	4.930E 01	9.670E 00
11.9	3.803E 04	3.534E 03	1.353E 02	1.870E 01	4.670E 00
28.1	3.927E 04	2.301E 03	6.400E 01	9.330E 00	6.700E-01
32.3	3.989E 04	2.652E 03	6.170E 01	3.300E 00	8.300E-01
36.6	4.730E 04	3.986E 03	1.290E 02	1.580E 01	4.170E 00
52.0	5.708E 04	4.304E 03	9.580E 01	5.830E 00	1.670E 00
60.5	6.442E 04	4.810E 03	1.042E 02	9.200E 00	3.330E 00
78.3	7.946E 04	6.299E 03	1.227E 02	1.200E 01	1.330E 00
94.3	8.162E 04	6.416E 03	1.133E 02	5.330E 00	2.670E 00
124.3	9.858E 04	8.247E 03	1.640E 02	1.730E 01	2.670E 00
137.2	9.487E 04	8.663E 03	3.233E 02	3.000E 01	6.670E 00
147.8	1.000E 05	8.403E 03	2.320E 02	1.830E 01	3.330E 00
150.7	9.857E 04	8.435E 03	1.700E 02	1.830E 01	8.330E 00
156.4	9.439E 04	7.900E 03	1.120E 02	5.000E 00	1.670E 00
172.8	1.274E 05	1.288E 04	2.870E 02	1.330E 01	1.670E 00
180.6	1.359E 05	1.406E 04	2.833E 02	1.170E 01	1.670E 00

SECTION VII

PISTON PUMP SPECIFICATION DEVELOPMENT PROJECT

PROJECT STAFF

Leonard E. Bensch, Project Manager

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FOREWORD

This report presents the results of an effort to develop a complete set of testing procedures and specifications for piston pumps which would be industrially acceptable and compatible with U.S. Army MERDC requirements. The effort has been directed toward pressure compensated piston pumps utilized for construction and earthmoving equipment. The primary emphasis of this year's work has centered about the development and verification of a contaminant sensitivity test for such pumps.

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CHAPTER I

INTRODUCTION

There is an obvious trend in the fluid power industry toward the greater use of both higher pressures and closed-center systems. The higher pressure permits the transfer of power with smaller flow rates and the accompanying smaller line sizes. However, higher pressure also requires either better structural materials in the components or greater wall thickness; thus, the actual net benefit is the subject of many debates. The use of high pressure hydraulic systems is forcing system designers toward those pumps primarily designed for such conditions – namely piston pumps.

The trend toward closed-center systems necessitates the use of a variable displacement pump. In the past, with open-center systems, pump flow remained constant, and the power which was consumed and transferred by the system was dependent upon the load. Open-center systems utilize a valve type which permits the flow that is not directed to the load to be bypassed back to the reservoir or to some other part of the system. When in the neutral position, the directional control valves of the open-center system permit the output of the pump (which only depends upon pump displacement and speed) to circulate back to the reservoir at a very low pressure. However, when the valve is actuated, the outlet pressure of the pump is dictated by the load.

In the closed-center system, the directional control valves are designed such that when they are in the neutral position all flow paths are blocked. Thus, it is necessary to utilize a pump which is capable of regulating its displacement in accordance with the demand of the operator. This is normally accomplished with a variable displacement pump which is equipped with pressure compensation. This means that the pump is automatically regulated to attempt to maintain a constant pressure until its maximum displacement is reached. There are two types of pumps which are fabricated in the variable displacement configuration and thus can be equipped

with the pressure compensator – vane and piston pumps. However, piston pumps are much more prevalent in modern high pressure hydraulic systems.

Since it is probable that much of the commercial construction machinery will incorporate both higher pressure and the closed-center designs in the near future, one of the projects of the MERDC-OSU Program was directed toward a piston pump specification effort. This project was intended to take maximum advantage of the experience gained through previous specification development efforts and the technical expertise and background of the fluid power industry. However, due to economic conditions, industrial participation was limited to correspondence and test pump support.

The investigation into the types of test procedures needed to specify a piston pump led to the conclusion that there are eight major evaluation methods required. The following is a listing of the titles of these test methods:

OSU-PC-1	**Method for Evaluating the Structural Integrity**
OSU-PC-2	**Method for Evaluating the Filling Characteristics**
OSU-PC-3	**Method for Evaluating the Steady-State Overall Efficiency Characteristics**
OSU-PC-4	**Method for Establishing the Durability**
OSU-PC-5	**Method for Evaluating the Low Speed and High Speed Performance**
OSU-PC-6	**Method for Evaluating the Low Temperature Performance**
OSU-PC-7	**Method for Establishing the Contaminant Sensitivity**
OSU-PC-8	**Method for Evaluating the Dynamic Response Characteristics**

Of these eight test procedures, four of them can be taken, with slight modification, from the previous MERDC-OSU effort, which was directed toward fixed displacement pumps. Specifically, the four procedures are OSU-PC-2, OSU-PC-4, OSU-PC-5, and OSU-PC-6. Of the remaining four, two are the subject of work being conducted by the pump committee of the National Fluid Power Association. In particular, this group is concerning itself with *"clean"*

oil performance characteristics of variable displacement piston pumps and thus have limited their efforts, at this time, to procedures designated OSU-PC-3 and OSU-PC-8. To avoid duplication of effort and still accomplish the objective of developing the necessary test procedures for evaluating variable displacement piston pumps, the MERDC-OSU team has concentrated its efforts on OSU-PC-1 and OSU-PC-7. In addition, since OSU-PC-1 is essentially a proof pressure test and therefore is relatively straightforward, the majority of the activity during the past contract year has been centered about the evaluation of contaminant sensitivity – OSU-PC-7.

In general, variable displacement piston pumps consist of two separate mechanisms. One mechanism, of course, is the pumping elements of the component, while the other is associated with the technique used to control the displacement of the pump. In addition, there are two major types of pumping mechanisms commonly found today in high pressure variable displacement piston pumps. The most abundant is the axial design where the motion of the reciprocating piston is parallel to the center line of the drive shaft. Somewhat less prevalent but still widely used is the radial configuration in which the reciprocating pistons move at right angles to the plane of the input shaft. However, the end user of the pump, the machine operator, is not concerned with the type of pumping mechanisms used nor the fact that two separate mechanisms are involved. His one and only concern is whether or not the pump outputs enough controlled flow to operate his machine properly.

Earlier attempts to evaluate the contaminant sensitivity of variable displacement pumps included two separate tests – one where the displacement varying mechanism was not involved and one where it was operational. The logic behind this approach was to evaluate not only the contaminant sensitivity of the pumping system but also to establish which of the mechanisms exhibited the most contaminant wear. In the MERDC-OSU effort, however, it was reasoned that the determination of the most sensitive element of the pump was not as important as the overall contaminant sensitivity. Thus, the contaminant sensitivity test for variable displacement piston pumps developed during the past year's effort involves the exposure of both the pumping and displacement varying mechanisms to the contaminant level simultaneously.

This report presents the results of the development effort on specification test procedures for variable displacement piston pumps. The primary objectives of this project were to determine the procedures needed, to assess the appropriateness of existing test procedures, to determine which applicable procedures were under consideration by standards making bodies, and to develop those necessary procedures which were unavailable. The major part of this effort was directed toward the development of a rigorous contaminant sensitivity test procedure. While all intrinsic procedures are discussed in this report, tests were conducted only to verify the contaminant sensitivity document. The results of these tests are presented along with an interpretation and conclusions.

CHAPTER II

INDUSTRIAL LIAISON

In previous specification and test procedure development efforts dealing with other hydraulic components, an effective and efficient approach was found in using the high talent expertise available from both end-item and component manufacturers to critique and guide the effort. In this approach, the MERDC-OSU project staff would utilize current procedures, verbal guidance from industrial sponsors as well as their own experience and creativity to compile a preliminary draft of each necessary test procedure. Two meetings would then be called to critique the effort. One of these meetings would consist of component representatives of end-item companies, while the other involved the component manufacturers' people. In this way, the MERDC-OSU staff would be given a clear picture of what was desired and also what could be expected. The industrial inputs from these meetings would be translated into appropriate terms; and, if necessary, the test procedure would be modified. When revised from these industrial inputs, this modified test procedure would be considered a test verification draft. Participating companies were asked to supply components which could be subjected to the test procedure. The results of these tests along with the revised test procedure would be presented to the industrial representatives at a second round of technical sessions. The inputs from this set of meetings would determine if additional development work was necessary.

In this piston pump specification effort, a similar approach was initiated. A letter was drafted explaining the objectives of the project and respectfully asking participation from those industrial companies which had some knowledge of the manufacturer and use of piston pumps. The letter was transmitted to 16 end-item company representatives who had supported such efforts. In addition, a similar letter was sent to 33 component manufacturers. The initial set of industrial meetings were set up for the 12-15 May of 1975 in Arlington, Virginia. The response from industry to this effort was quite disappointing. Of the 16 end-item companies

contacted, only three indicated that they would have technical representatives at the meeting. Furthermore, only four the 33 component companies said that they could attend.

The vast majority of the industrial companies who responded to the request to participate in this effort strongly cited economic conditions as their reason for not attending this most important initial meeting. While they urged the continuation of the activity and asked to be kept informed of the progress and results, these companies indicated that their participation would have to be limited to correspondence until some undefined time later in the year. Based upon these inputs, the decision was made to cancel the Arlington meeting and rely upon written and telephone communications as well as company visits by project staff to obtain industrial guidance.

Trips were made to nine of the industrial companies to discuss the piston pump specification development effort. These companies were all manufacturers of end-item machinery and had expressed an interest in the test procedure development. The general concensus of the conversations during these visits as well as during other communications was that the most important test procedure needed for evaluating pressure compensated piston pumps was associated with contaminant sensitivity. Therefore, a concerted effort and a major portion of the project activity was directed toward the development and verification of a contaminant sensitivity test for pressure compensated piston pumps.

The pressure compensated piston pump contaminant sensitivity test procedure was written based upon experience gained during the development and verification of the contaminant sensitivity test procedure for fixed displacement hydraulic pumps. In addition, the industrial inputs were included where appropriate. The preliminary procedure was then submitted to interested industrial representatives to obtain any additional inputs and opinions which they deemed applicable. Along with the preliminary test procedure, a request was transmitted asking the participating companies to send pressure compensated piston pumps to be subjected to the new test. In all, eight pumps were received from four industrial companies. These pumps were tested per the test procedure, and the results and analysis are contained in this report.

CHAPTER III

PROCEDURE DEVELOPMENT

A pressure compensated piston pump is defined to achieve a variable delivery to match the needs of the system in which it operates. To accomplish this, the pump actually consists of two separate systems, usually located in the same housing. Therefore, it is reasonable to consider testing each system separately. This would be ideal from a component manufacturer's viewpoint because the data from the two tests would pinpoint problem areas more explicitly. From the Army's standpoint, a pressure compensated piston pump is an integral unit and should be evaluated as such. This would certainly save testing time, since it would not be necessary to conduct two tests. However, it may be difficult to determine the performance characteristics of the pumping system exclusive of the compensating mechanism and vice versa.

All of the test procedures included in this report are designed to test the pressure compensated piston pumps as an integral unit. In addition, the two procedures under study by the NFPA Pump Committee are also intended to evaluate the entire pump assembly. All eight of the test procedures recommended for the evaluation of pressure compensated piston pumps are discussed separately in this section of this report in order to outline the basis for each. The results of the verification effort on the contaminant sensitivity procedure are included in a later section.

OSU-PC-1 METHOD FOR EVALUATING THE STRUCTURAL INTEGRITY

The purpose of this test is to evaluate the physical condition of the pump after exposure to an above normal operating condition. In a fixed displacement pump, this is easily accomplished by restricting the output flow until the desired pressure is obtained. It is common to subject such pumps to 130% of rated pressure in this test. With a pressure compensated piston

pump, however, the flow can be completely blocked without achieving more than rated standby pressure. To overcome this factor, a high pressure, low flow auxiliary pump is used to attain a pressure greater than rated.

In conducting the test, the pump is operated at rated speed, 50°C inlet oil temperature, and atmospheric inlet pressure. As shown in the figure which accompanies the test procedure in Appendix A, the output of the auxiliary pump is directed into the outlet line of the test pump. When the load valve is partially closed, the test pump will go to zero flow, but the auxiliary pump will supply sufficient flow to reach a value of 130% of the rated standby or zero flow pressure of the test pump. A structural failure as evidenced by external leakage is the failure criterion for this test.

OSU-PC-2 METHOD FOR EVALUATING THE FILLING CHARACTERISTICS

In designing any hydraulic system, it is mandatory that the inlet line to the pump be fabricated in such a manner that the flow is not unduly restricted. If an adequate inlet is not provided, the pump will exhibit what is commonly called *cavitation*, where the pumping chamber will not be completely filled with hydraulic oil. This condition can cause the pump to emit an excessive noise level and can seriously reduce the service life. In this test procedure, *filling characteristic* is defined as a feature of a fluid power pump which indicates the void volume within the pumping chambers at specified operating conditions. In a pressure compensated piston pump, the filling characteristics are especially important because a cavitating pump will exhibit erratic output flow and pressure, which will be sensed by the compensator causing an unstable situation.

During the test, the pump is operated at a discharge pressure of 35 bars, inlet pressure of atmospheric $\pm$ 25 mm Hg and at manufacturer's recommended inlet temperature. The pump speed is varied from 600 RPM to manufacturer's rated speed, and the output flow is

recorded to establish the normal speed versus flow curve of the pump. Then, the inlet pressure is reduced to atmospheric minus 25 mm Hg, while the pump speed is maintained at the rated value. The flow rate at the restricted inlet condition is recorded as an indication of the filling characteristics of the test pump. Failure to emit a specified flow rate at the test inlet pressure is considered as the appraisal criterion for the test. The test procedure is given in Appendix A.

OSU-PC-3 METHOD FOR EVALUATING THE STEADY-STATE OVERALL EFFICIENCY CHARACTERISTICS

The steady-state performance of a pressure compensated piston pump is an important consideration in procurement. In general, the steady state characteristics of such a pump can be typified, as shown in Fig. 3-1. The pump will remain at full design displacement until the pressure is increased sufficiently. As the pressure is increased further, the displacement and, hence, the output flow will reduce. The shape of the curve as shown in Fig. 3-1 will be reflected in the performance exhibited by a system using the pump.

The Pump and Motor Committee of the National Fluid Power Association (NFPA) is currently developing a test procedure which will provide the necessary information to evaluate the steady-state performance of a pressure compensated pump. The document being generated by this group includes both the steady-state performance test and the transient or dynamic response characteristic. The MERDC-OSU project staff recommend that these two evaluation tests be separated, since they essentially evaluate the unit under entirely different conditions. Therefore, the two tests carry different numbers (OSU-PC-3 and OSU-PC-8) and will be discussed separately in this report. The NFPA document entitled *"Method of Testing and Presenting Basic Performance Data for Positive Displacement, Variable Volume, Pressure Compensated Hydraulic Fluid Power Pumps"* is currently at the first draft stage in the Pump Committee (Project Group T3.9.9.20).

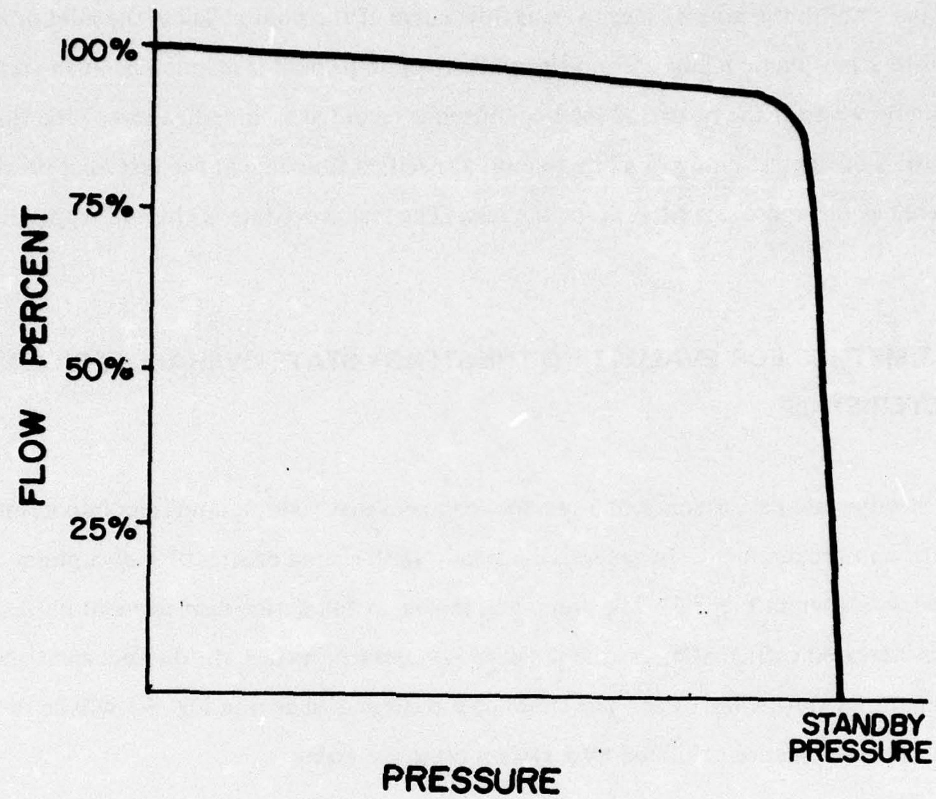


Fig. 3-1. *Typical Steady-State Performance of a Pressure Compensated Piston Pump.*

Essentially, the test assesses the volumetric displacement of the unit by operating at 1000 RPM, 6.3 bar output pressure, manufacturer's recommended inlet pressure, and recording the outlet flow at these conditions. Then, with the compensator blocked or set high enough so as to present no interference,the pump is tested per the NFPA procedure for fixed displacement pumps (T3.9.17). The output flow and power input at various outlet pressures are measured. From this information, the overall efficiency and volumetric displacement of the pump mechanism can be evaluated.

The steady-state characteristics are assessed using three different shaft speeds and three different compensator settings at each shaft speed. The test procedure calls for output flow versus output pressure to be recorded for both increasing and decreasing values of output pressure. The test schedule as laid out by NFPA calls for 30 data points to be taken in evaluating the steady-state pressure compensator characteristics. Since data points are recorded with increasing and decreasing pressure, sufficient information is available to evaluate the hysteresis of the compensating mechanism.

OSU-PC-4 METHOD FOR ESTABLISHING THE DURABILITY

Service life of hydraulic components is determined by two factors. One factor is the contamination level of the system and is considered in OSU-PC-7. The other aspect is associated with the fatigue characteristics of the materials of the component and is normally appraised by a test which subjects the component to a cyclic pressure. The peak value of pressure wave for the normal durability test is 115% of rated pressure. However, due to the manner of operation of a pressure compensated pump, the standby or zero flow pressure cannot be conveniently exceeded.

The durability test proposed for pressure compensated piston pumps is included in the appendix. In this test, the system fluid is maintained at a low contamination level to prevent the possibility of contaminant wear. The test pump is operated at the manufacturer's rated speed and recommended inlet oil temperature. With the inlet pressure maintained at atmospheric, the outlet pressure is cycled from 5 to 100% of standby or dead-head pressure, which of course means that the flow rate also cycles from full to zero. The cycle rate is 60 cycles/minute with at least 1/2 of the cycle time to be at 100% pressure. The test specified this cycle to be repeated 500,000 times.

In addition to the cycle test, the procedure also calls for a constant pressure test for an equivalent of 50 hours. The value of the pressure during this test is 75% of standby or dead-head

pressure. The logic behind this portion of the test lies in the fact that during constant pressure operation the bearings, shaft, etc. are continuously loaded in one direction. Thus, the supporting and rotating elements of the pump will receive more stress cycles at constant pressure than during the variable pressure requirement.

OSU-PC-5 METHOD FOR EVALUATING THE LOW SPEED AND HIGH SPEED PERFORMANCE

In mobile construction machinery, the pump of a hydraulic system is normally driven by a reciprocating engine. Since this type of engine does not operate at constant speeds, it is necessary to evaluate the characteristics of the hydraulic pump when it is subjected both to low and high speed operation. To accomplish such an evaluation the pump is tested at a discharge pressure of 75% of standby and at speeds of 600 RPM and 115% of rated speed. Both the pump speed and discharge pressure are recorded versus time. The failure criterion is any erratic behavior or external leakage.

OSU-PC-6 METHOD FOR EVALUATING THE LOW TEMPERATURE PERFORMANCE

When hydraulic systems are required to operate in cold climates, the low temperature performance characteristics of the pump must be considered. This test procedure is designed to verify the ability of a pressure compensated piston pump to withstand low temperature operation and perform satisfactorily at specified conditions of speed and discharge pressure. In order to accomplish this verification, the test procedure requires the use of a cold room capable of maintaining an air temperature of -30°C. Since the hydraulic test system as well as the pressure compensated pump must be lowered to this temperature, the volume of the test system is limited to one-half the output flow of the pump. This helps to reduce the size of the cold room necessary for the test.

The pump and hydraulic system are subjected to the -30° C environment for a period of 12 hours prior to testing. The pump is then operated according to a temperature schedule developed through industrial guidance for fixed displacement pumps. The complete test procedure is included in Appendix A of this report. To document the test condition, pressure, temperature, and speed are all recorded with respect to time. Erratic operation and external leakage are used as test criteria in this procedure.

OSU-PC-7 METHOD FOR ESTABLISHING THE CONTAMINANT SENSITIVITY

The life of a pressure compensated piston pump is considered to be terminated when it no longer delivers a specified flow rate at a given shaft speed, discharge pressure, and fluid condition or when the standby pressure changes by a specified amount. The pump may reach the terminal state due to catastrophic (mechanical interference or material overstress) failure or by the cumulative effect of wear processes. The wear rate within a pressure compensated piston pump is proportional to the contamination level of the hydraulic fluid exposed to the internal surfaces of the pump. This test procedure is designed to evaluate the contaminant wear characteristics of such pumps.

Since a pressure compensated pump represents two different mechanisms, the test is conducted in such a way as to evaluate both the pumping mechanism and the compensating valves. Any contaminant wear of the surfaces which form the critical clearance spaces (leakage paths) of the pressure compensated piston pump will be accompanied by a measurable degradation in its delivered flow rate. In addition, any critical wear associated with the compensating parts of the pump will be reflected by a change in the standby or zero flow pressure measurement. Based upon these considerations, pressure and flow degradation ratios are used in this test to establish the contaminant sensitivity of a pressure compensated piston pump. Both of these parameters can be determined from the results of only one contaminant test on the pump.

Essentially, the test consists of operating the pump at rated speed. The volume of the system is adjusted to be numerically equal to one-fourth the flow rate measured at 67% of standby pressure. The pressure of the test pump is adjusted to achieve a flow rate equal to 0.5 of the rated flow (measured at the 67% standby pressure point). The pump is subjected to a 300 mg/litre level of various size ranges of contaminant classified from AC Fine Test Dust (0-5, 10, 20, 30, 40, 50, 60, 70, and 80 micrometres). The test procedure specifies that the flow be recorded at 67% of standby and the actual value of standby pressure be recorded after each contaminant exposure. However, during the verification test conducted per this procedure, sufficient flow-pressure data were recorded after each injection to define the entire curve.

The next section of this report contains the results of the verification tests run on the pressure compensated piston pumps supplied by participating industrial companies. In addition, the detailed contaminant sensitivity test procedure is included in Appendix A of this report.

OSU-PC-8 METHOD FOR EVALUATING THE DYNAMIC RESPONSE CHARACTERISTICS

This test procedure is a second part of the effort by the Pump Committee of the National Fluid Power Association. Since a pressure compensated pump is expected to maintain a relatively constant output pressure when subjected to rapidly changing loads, its transient performance is of utmost importance in system response. To evaluate the dynamic performance of the test pump, a manual shut-off valve and a rapid shut-off valve are installed in the outlet line from the pump. The manual valve is adjusted to achieve an outlet pressure of 75% of standby or dead-head pressure. Then, the rapid shut-off valve is cycled on and off while a pressure versus time recording is made.

From the pressure data taken during the test, it is possible to assess the response time, recovery time, the pressure overshoot, the pressure undershoot, and the decay time associated

with the pump. The response time and the pressure overshoot of a pressure compensated pump are defined by the NFPA document as shown in Fig. 3-2. Fig. 3-3 shows the definition of the recovery time and the pressure undershoot while Fig. 3-4 illustrates the meaning of decay time. All of these values are determined in a clean oil environment to prevent interference by contaminants.

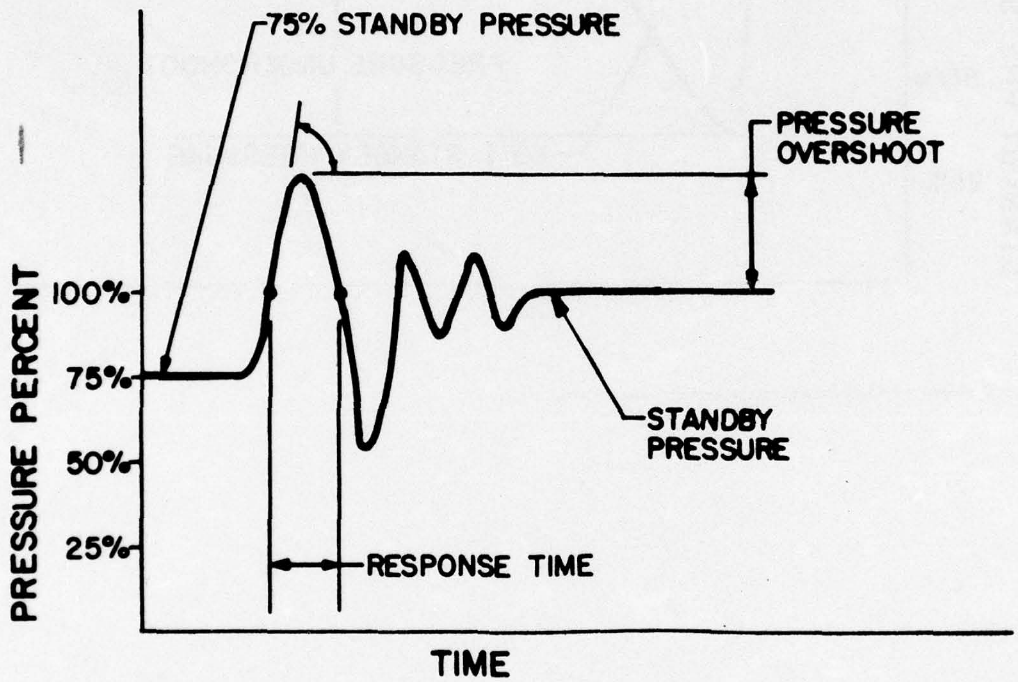


Fig. 3-2. Response of a Pressure Compensated Pump.

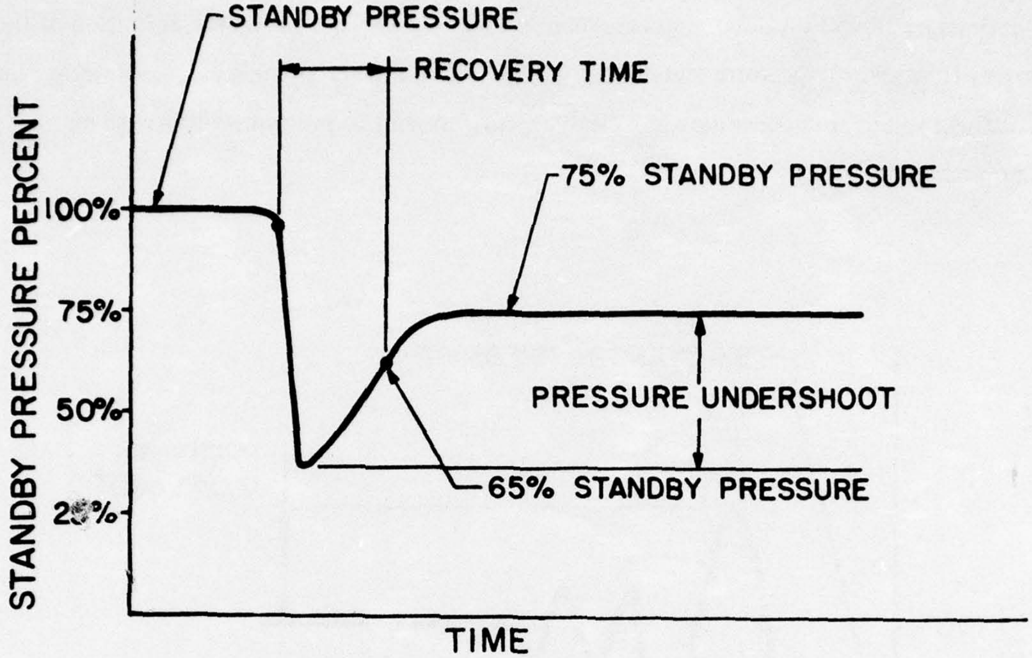


Fig. 3-3. Recovery Time of a Pressure Compensated Pump.

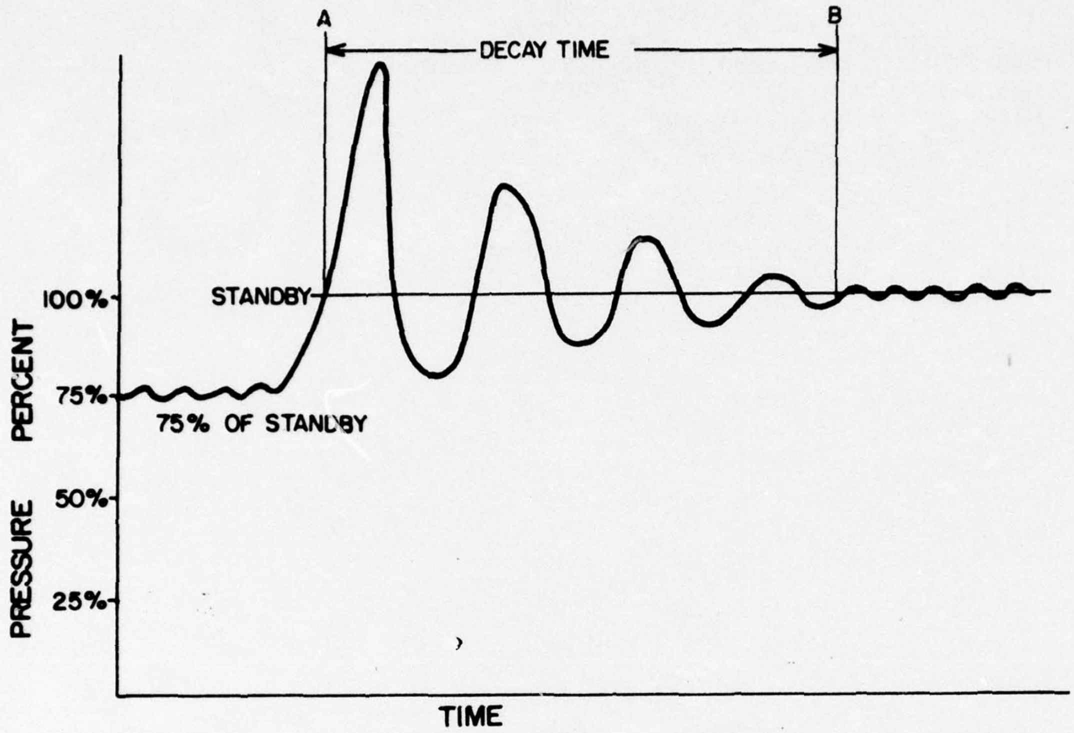


Fig. 3-4. Pressure Transient Decay of a Pressure Compensated Pump.

CHAPTER IV

VERIFICATION TESTS

Based upon the industrial guidance obtained during this effort, the contaminant sensitivity test was considered the most critical and undeveloped of the eight test procedures for pressure compensated piston pumps. At the beginning of this activity, the results from only three contaminant tests had been reported for this type of pump [1]. These three pumps had different pumping displacements but were all of the same design. Two tests were conducted on each pump – one with the compensator blocked and one with the compensator active. These results were not completely satisfactory, and it was obvious that more development work on the procedure was necessary. Therefore, the verification testing accomplished during this activity was centered about contaminant sensitivity.

In all, eight pressure compensated piston pumps were received from four different participants. Two of the participating companies furnished two pumps each – one furnished three pumps and one supplied one pump – making a total of eight. For identification purposes, the pumps will be labeled A1, A2, B1, B2, B3, C1, C2, and D, where the letter indicates the supplier, and the number refers to additional pumps of identical design from the same company.

Figs. 4-1 through 4-5 graphically illustrate the flow pressure data obtained during the contaminant tests on five typical pumps. During the testing phase, it was found necessary to modify the test procedure slightly from the first draft. Initially, the procedure specified that the test system volume should be numerically equal to one-eight of the pump flow measured at 75% of standby pressure. During testing, it was discovered that this volume could not be achieved with all pumps; therefore, the volume requirement was changed to be numerically equal to one-fourth the rated flow. In addition, it was found that some pumps began compensating before 75% of standby pressure, and it was necessary to take the flow reading at 67% of the dead-head pressure.

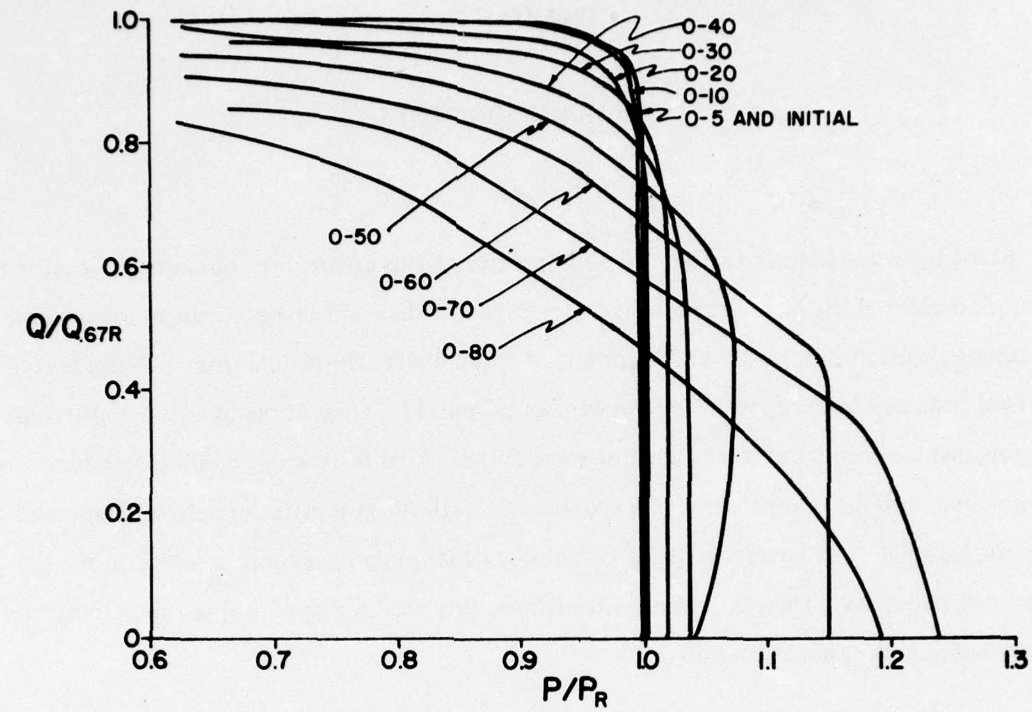


Fig. 4-1. Flow Versus Pressure Curves from Pump A1.

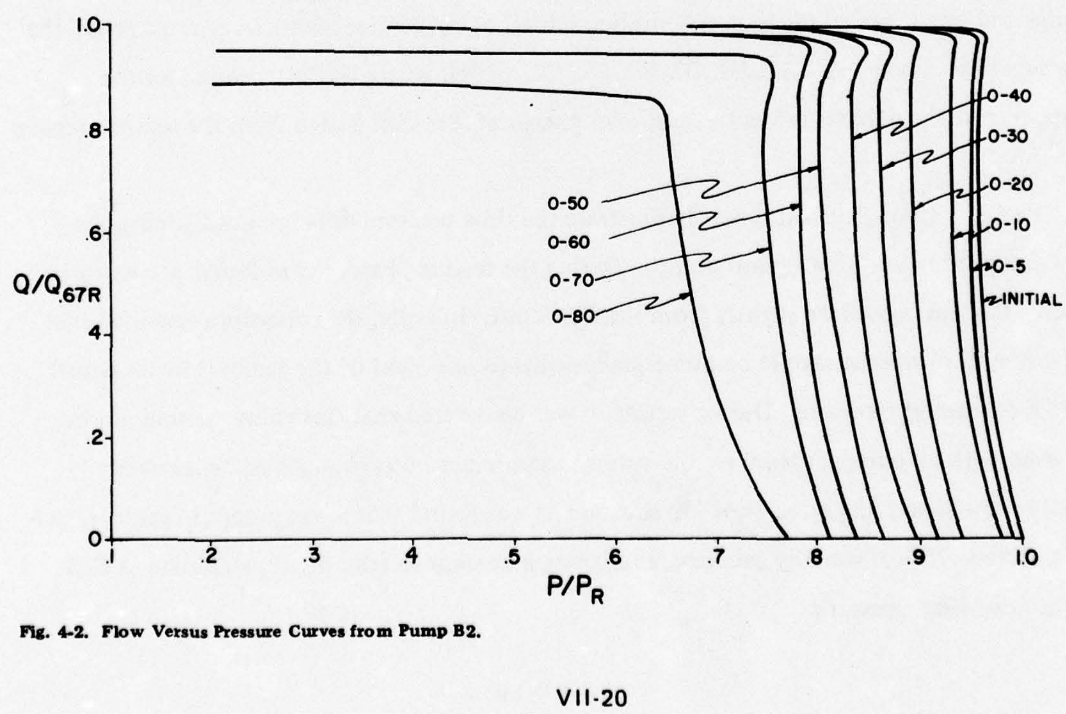


Fig. 4-2. Flow Versus Pressure Curves from Pump B2.

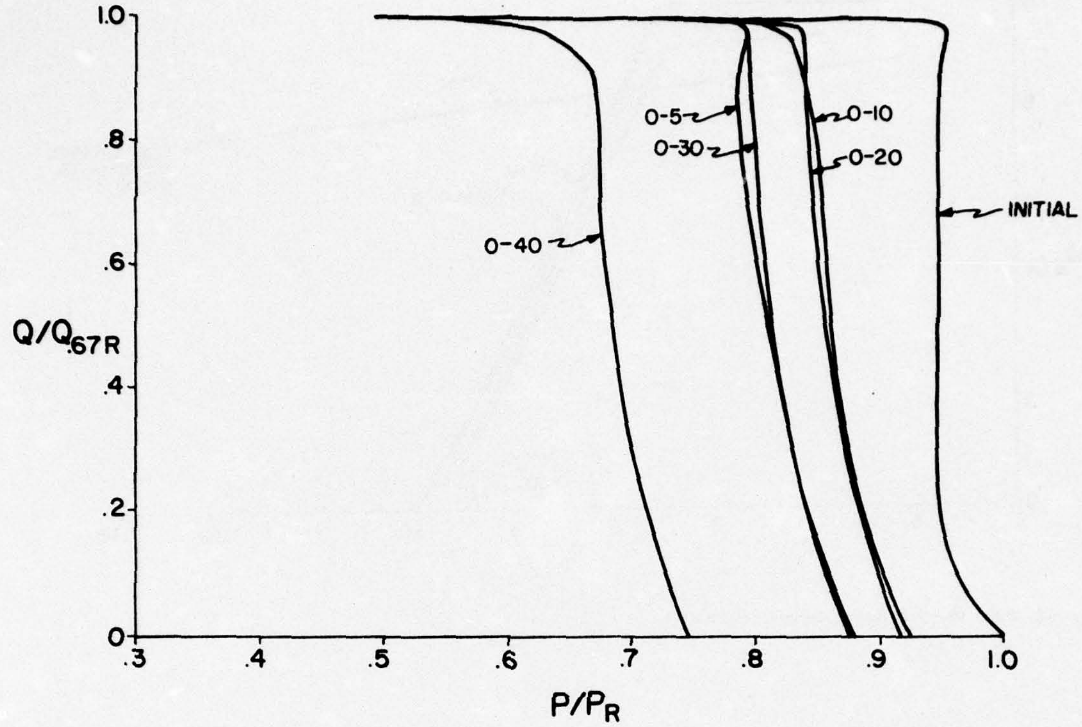


Fig. 4-3. Flow Versus Pressure Curves from Pump B3.

Fig. 4-1 shows the flow ratio versus the pressure ratio for Pump A1. It should be noted from this figure that, while the flow ratio decreased from contaminant exposure, the pressure ratio actually increased. The results of Pump B2 are shown in Fig. 4-2, and those of Pump B3 are shown in Fig. 4-3. The difference between these two tests is that Pump B2 was run at the one-eighth volume, while Pump B3 was tested at the one-fourth volume. Pumps B2 and B1 were the only pumps tested with the system volume numerically equal to one-eighth of rated pump flow. It can be seen by comparing the test results from Pumps B2 and B3 that the test is more severe with the larger volume. This is because there are more total particles present when the larger volume is used, even though the contamination level is the same (300 mg/litre).

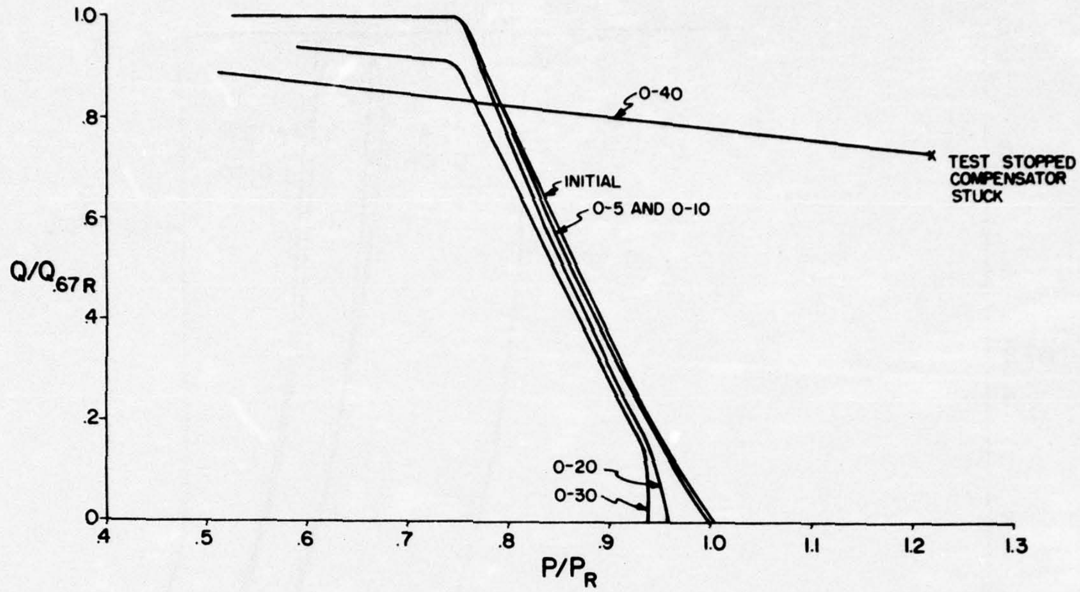


Fig. 4-4. Flow Versus Pressure Curve from Pump C2.

Fig. 4-4 graphically shows the results from the contaminant sensitivity test on Pump C2. It should be noted from the curves in Fig. 4-4 that this pump began compensating below 75% of the standby pressure. This result necessitated a change in the test procedure to require that the flow after contaminant exposure be measured at 67% instead of the 75% point, as originally stated. In addition, it can be seen from Fig. 4-4 that, when the test reached the 0-40 micrometer injection, the compensator apparently ceased to function, and the pump acted as if it were of the fixed displacement type. Fig. 4-5 shows the flow versus pressure results from Pump D. The compensator on this pump apparently malfunctioned on the 0-50 micrometre injection. However, contrary to the pump shown in Fig. 4-4, Pump D exhibited a loss of flow when the malfunction occurred.

Even though, in these verification tests, the entire flow-pressure curve of the test pump was evaluated for each contaminant exposure, it is felt that the contaminant sensitivity of a pressure compensated pump can be assessed from two characteristics. The flow degradation

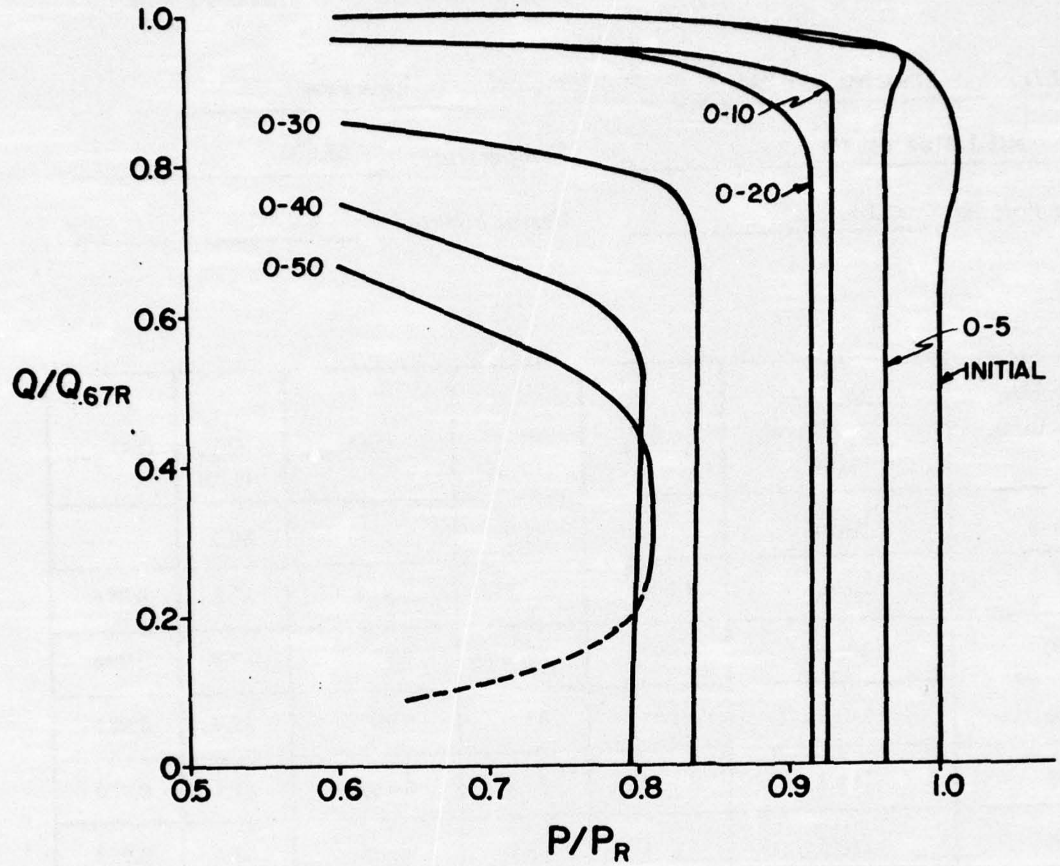


Fig. 4-5. Flow Versus Pressure Curves from Pump D.

ratio calculated from data taken at a pressure of 67% of the measured dead-head pressure will provide information relative to deterioration of the pumping mechanism. Also, the pressure ratio defined as the standby pressure will indicate a change in the compensator. Tables 4-1 through 4-8 are summaries of the data as required by the test procedure for all eight pumps tested. Since the first draft of the test procedure specified a flow reading at 75% of standby pressure and later revisions changed this to 67%, both flow readings are given in the tables. The nomenclature used in these tables is as follows:

TABLE 4-1

SUMMARY OF DATA FOR PRESSURE COMPENSATED PISTON PUMP

Pump I.D.: FPRC NO. 255 "A1" *Speed:* 2400 RPM

Fluid: MIL-L-2104 CL. 10 *Temperature:* 65.5°C

System Volume: 22.1 ℓ *Grams Injected:* 6.6/INJ.

Particle Size Range (μM)	Pressure @ Zero Flow (bar)	P/P_z	$Q_{.75}$ (LPM)	$Q/Q_{.75R}$	$Q_{.67}$ (LPM)	$Q/Q_{.67R}$
Rated	209.0	----	88.2	----	88.2	----
0-5	209.0	1.000	87.8	0.996	87.8	0.996
0-10	208.3	0.997	87.4	0.991	87.8	0.996
0-20	210.3	1.007	87.1	0.987	87.4	0.991
0-30	212.4	1.017	84.8	0.961	87.1	0.970
0-40	216.6	1.036	83.7	0.948	85.6	0.953
0-50	217.2	1.040	81.0	0.918	84.0	0.927
0-60	240.0	1.149	78.0	0.884	81.8	0.893
0-70	258.6	1.238	74.2	0.841	78.7	0.850
0-80	248.3	1.188	68.1	0.773	70.8	0.803

TABLE 4-2

SUMMARY OF DATA FOR PRESSURE COMPENSATED PISTON PUMP

Pump I.D.: FPRC NO. 256 "A2" *Speed:* 2400 RPM

Fluid: MIL-L-2104 CL. 10 *Temperature:* 65.5°C

System Volume: 21.6 ℓ *Grams Injected:* 6.5/INJ.

Particle Size Range (μM)	Pressure @ Zero Flow (bar)	P/P_r	$Q_{.75}$ (LPM)	$Q/Q_{.75R}$	$Q_{.67}$ (LPM)	$Q/Q_{.67R}$
Rated	206.9	----	86.3	---	86.7	----
0-5	205.2	0.992	86.3	1.000	86.7	1.000
0-10	205.5	0.993	85.2	0.987	86.7	1.000
0-20	205.9	0.995	84.8	0.982	86.3	0.996
0-30	248.3	1.200	81.4	0.943	83.3	0.961
0-40	213.8	1.033	80.3	0.930	81.8	0.943
0-50	220.3	1.065	77.6	0.899	78.7	0.908
0-60	231.7	1.120	76.1	0.882	77.6	0.895
0-70	233.1	1.127	73.1	0.846	75.7	0.873
0-80	232.8	1.125	65.9	0.763	69.7	0.803

TABLE 4-3

SUMMARY OF DATA FOR PRESSURE COMPENSATED PISTON PUMP

Pump I.D.: FPRC NO. 253 "B1" *Speed:* 2200 RPM

Fluid: MIL-L-2104 CL. 10 *Temperature:* 65.5°C

System Volume: 12.7ℓ *Grams Injected:* 3.8/INJ.

Particle Size Range (μM)	Pressure @ Zero Flow (bar)	P/P_r	$Q_{.75}$ (LPM)	$Q/Q_{.75R}$	$Q_{.67}$ (LPM)	$Q/Q_{.67R}$
Rated	165.5	---	101.8	---	102.2	----
0-5	163.8	0.990	101.8	1.000	102.2	1.000
0-10	16.0	0.969	101.8	1.000	102.2	1.000
0-20	156.9	0.948	101.8	1.000	102.2	1.000
0-30	153.4	0.927	101.8	1.000	102.2	1.000
0-40	148.3	0.896	101.8	1.000	102.2	1.000
0-50	144.8	0.875	101.8	1.000	102.2	1.000
0-60	140.0	0.846	99.6	0.978	100.3	0.981
0-70	137.6	0.831	99.2	0.974	99.2	0.970
0-80	131.0	0.792	98.4	0.967	98.4	0.963

TABLE 4-4

SUMMARY OF DATA FOR PRESSURE COMPENSATED PISTON PUMP

Pump I.D.: FPRC NO. 254 "B2" *Speed:* 2200 RPM

Fluid: MIL-L-2104 CL. 10 *Temperature:* 65.5°C

System Volume: 12.5 ℓ *Grams Injected:* 3.8/INJ.

Particle Size Range (μM)	Pressure @ Zero Flow (bar)	P/P_r	$Q_{.75}$ (LPM)	$Q/Q_{.75R}$	$Q_{.67}$ (LPM)	$Q/Q_{.67R}$
Rated	165.5	---	100.3	----	100.7	----
0-5	163.4	0.988	100.3	1.000	100.7	1.000
0-10	162.1	0.979	100.3	1.000	100.7	1.000
0-20	155.5	0.940	100.3	1.000	100.7	1.000
0-30	150.7	0.910	100.3	1.000	100.7	1.000
0-40	147.2	0.890	100.3	1.000	100.7	1.000
0-50	142.4	0.860	99.2	0.989	99.6	0.989
0-60	138.6	0.838	98.4	0.981	98.8	0.981
0-70	133.8	0.808	93.5	0.932	95.0	0.944
0-80	127.2	0.769	6.8	0.068	66.2	0.658

TABLE 4-5

SUMMARY OF DATA FOR PRESSURE COMPENSATED PISTON PUMP

Pump I.D.: FPRC NO. 260 "B3" *Speed:* 2200 RPM

Fluid: MIL-L-2104 CL. 10 *Temperature:* 65.5 GPM

System Volume: 25.5 ℓ *Grams Injected:* 7.6/INJ.

Particle Size Range (μM)	Pressure @ Zero Flow (bar)	P/P_r	$Q_{.75}$ (LPM)	$Q/Q_{.75R}$	$Q_{.67}$ (LPM)	$Q/Q_{.67R}$
Rated	165.5	---	101.8	----	102.2	----
0-5	144.8	0.875	101.8	1.000	102.2	1.000
0-10	152.4	0.921	101.8	1.000	102.2	1.000
0-20	151.7	0.917	101.8	1.000	102.2	1.000
0-30	144.8	0.875	101.8	1.000	102.2	1.000
0-40	122.8	0.742	0.0	0.000	81.8	0.800
0-50						
0-60						
0-70						
0-80						

TABLE 4-6

SUMMARY OF DATA FOR PRESSURE COMPENSATED PISTON PUMP

Pump I.D.: FPRC NO. 258 "C1" *Speed:* 2300 RPM

Fluid: MIL-L-2104 CL.10 *Temperature:* 65.5°C

System Volume: 49.7 ℓ *Grams Injected:* 14.9/INJ.

Particle Size Range (μM)	Pressure @ Zero Flow (bar)	P/P_r	$Q_{.75}$ (LPM)	$Q/Q_{.75R}$	$Q_{.67}$ (LPM)	$Q/Q_{.67R}$
Rated	173.1	---	198.7	----	199.5	----
0-5	174.8	1.010	197.6	0.994	199.1	0.998
0-10	173.4	1.002	197.2	0.992	199.1	0.998
0-20	172.4	0.996	193.4	0.973	195.7	0.981
0-30	*		171.5	0.863	179.4	0.899
0-40						
0-50						
0-60						
0-70						
0-80						

*Compensator stuck; maximum pressure >> 200 bars.

TABLE 4-7

SUMMARY OF DATA FOR PRESSURE COMPENSATED PISTON PUMP

Pump I.D.: FPRC NO. 259 "C2" *Speed:* 2300 RPM

Fluid: MIL-L-2104 CL. 10 *Temperature:* 65.5°C

System Volume: 49.5 ℓ *Grams Injected:* 14.8 g

Particle Size Range (μM)	Pressure @ Zero Flow (bar)	P/P_r	$Q_{.75}$ (LPM)	$Q/Q_{.75R}$	$Q_{.67}$ (LPM)	$Q/Q_{.67R}$
Rated	175.9	--	198.0	----	198.7	---
0-5	175.9	1.000	198.0	1.000	198.7	1.000
0-10	175.9	1.000	197.6	0.998	198.0	0.996
0-20	167.9	0.955	197.6	0.998	198.0	0.996
0-30	164.8	0.937	179.8	0.908	184.3	0.928
0-40	*		166.2	0.839	169.6	0.853
0-50						
0-60						
0-70						
0-80						

*Compensator stuck; maximum pressure >> 200 bars.

TABLE 4-8

SUMMARY OF DATA FOR PRESSURE COMPENSATED PISTON PUMP

Pump I.D.: FPRC NO. 257 "D" *Speed:* 2400 RPM

Fluid: MIL-L-2104 CL. 10 *Temperature:* 65.5°C

System Volume: 21.6 ℓ *Grams Injected:* 6.5/INJ.

Particle Size Range (μM)	Pressure @ Zero Flow (bar)	P/P_r	$Q_{.75}$ (LPM)	$Q/Q_{.75R}$	$Q_{.67}$ (LPM)	$Q/Q_{.67R}$
Rated	170.3	---	86.3	---	86.3	----
0-5	163.8	0.962	85.9	0.996	86.3	1.000
0-10	157.6	0.925	82.9	0.961	83.7	0.969
0-20	151.7	0.891	82.1	0.952	83.3	0.965
0-30	142.4	0.836	70.0	0.811	72.3	0.838
0-40	135.2	0.794	55.6	0.645	59.4	0.689
0-50	13.8	0.081	46.2	0.535	51.9	0.601
0-60						
0-70						
0-80						

PRESSURE @ ZERO FLOW = STANDBY PRESSURE MEASURED INITIALLY AND AFTER EACH CONTAMINANT INJECTION

P/P_r = the standby pressure after each contaminant exposure divided by the measured rated standby pressure

$Q_{.75}$ = flow rate measured at 75% of standby pressure

$Q_{.67}$ = flow rate measured at 67% of standby pressure

$Q/Q_{.75}$ and $Q/Q_{.67}$ = FLOW DEGRADATION RATIO FROM THE FLOW INFORMATION OBTAINED AT 75% AND 67% OF STANDBY PRESSURE RESPECTIVELY

The pressure ratio data shown in Table 4-1 through 4-8 are graphically illustrated in Fig. 4-6. In viewing Fig. 4-6, it should be noted that A1 and A2 are *"identical"* pumps tested under the same conditions and show excellent repeatability, as do C1 and C2 and B1 and B2. The volume change explained previously was made after testing pumps B1 and B2. Therefore, while B2 is the same pump as B1 and B2, the test conditions are different. Pumps A1, A2, B3, C1, C2, and D were all tested at the higher volume requirement. It was decided to conduct the test on Pump B3 to demonstrate the effect of the volume change. It should be noted that the pressure ratio on Pumps C1 and C2 gets very large at 0-30 and 0-40 respectively. This is because both of these pumps failed to limit the pressure at all when a compensator malfunction occurred.

The flow degradation ratio versus contaminant size range injected is shown in Figs. 4-7 and 4-8 for 75% standby and 67% standby pressure respectively. Since the rated value of flow which was used to normalize the data was actually measured during the test, there is little difference between these two figures. However, in some cases, the 75% point is very close to the pressure where the compensator becomes active; thus, the flow measurement at 67% of standby is recommended.

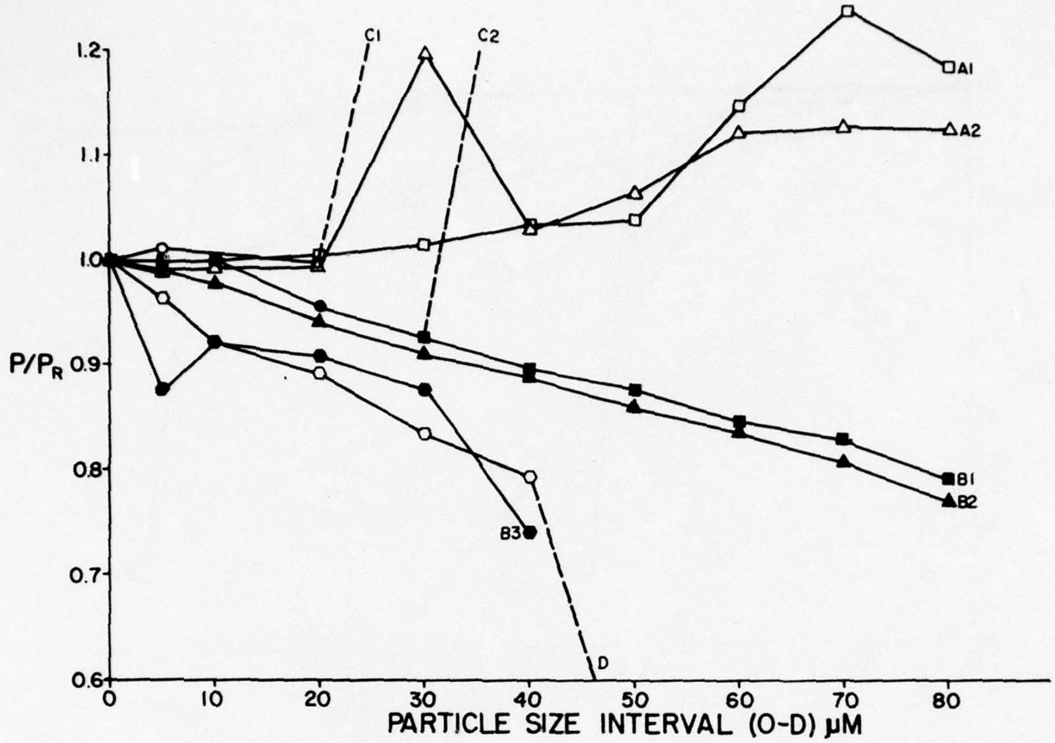


Fig. 4-6. Pressure Ratio Versus Size Range Injected.

The results of a contaminant sensitivity test on a pressure compensated piston pump do not need special interpretation in order to gain an appreciation for the information. The fact that the pressure ratio or flow ratio changes as the pump is exposed to various contaminant environments is sufficient to evaluate the deleterious influence of particulate contaminants. However, to satisfy the objectives of different factions of the fluid power industry, the Fluid Power Research Center has developed a method for describing the contaminant wear tolerance [2, 3].

Since there are two performance parameters – flow ratio and pressure ratio – there are three contaminant tolerance profiles which can be calculated from the test data. One profile is based upon the degradation in flow measured at 67% of standby pressure and can be determined as detailed in Ref. [2]. The second tolerance profile is based upon the absolute value of the standby pressure changes calculated as given in Ref. [3]. The third profile, of course,

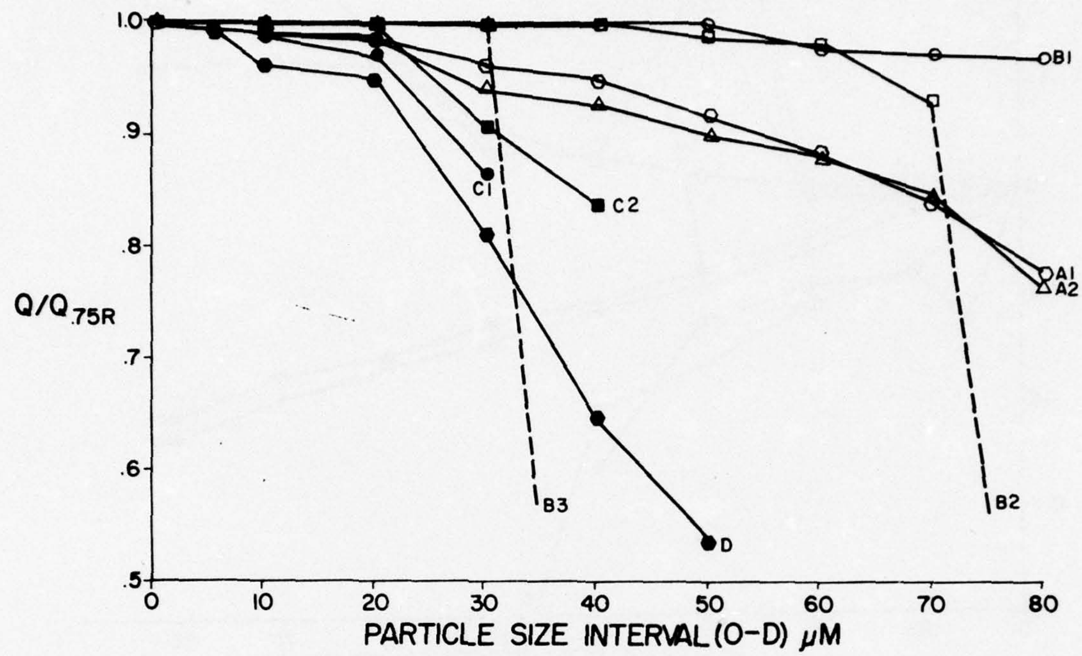


Fig. 4-7. Flow Degradation Ratio Vs. Size Range Injected at 75% of Initial Standby Pressure.

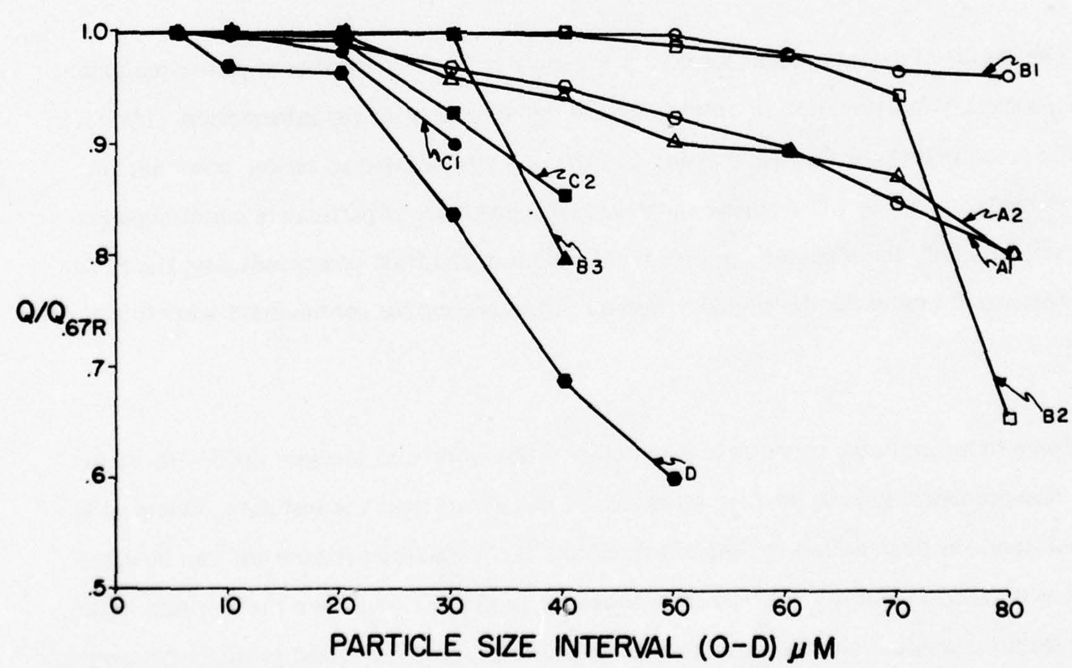


Fig. 4-8. Flow Degradation Ratio Versus Size Range Injected at 67% of Initial Standby Pressure.

is a composite of the other two and probably describes the pump most accurately. The 1000 hour contaminant tolerance profiles are shown in Figs. 4-9 through 4-13 for Pumps A1, B2, B3, C2, and D.

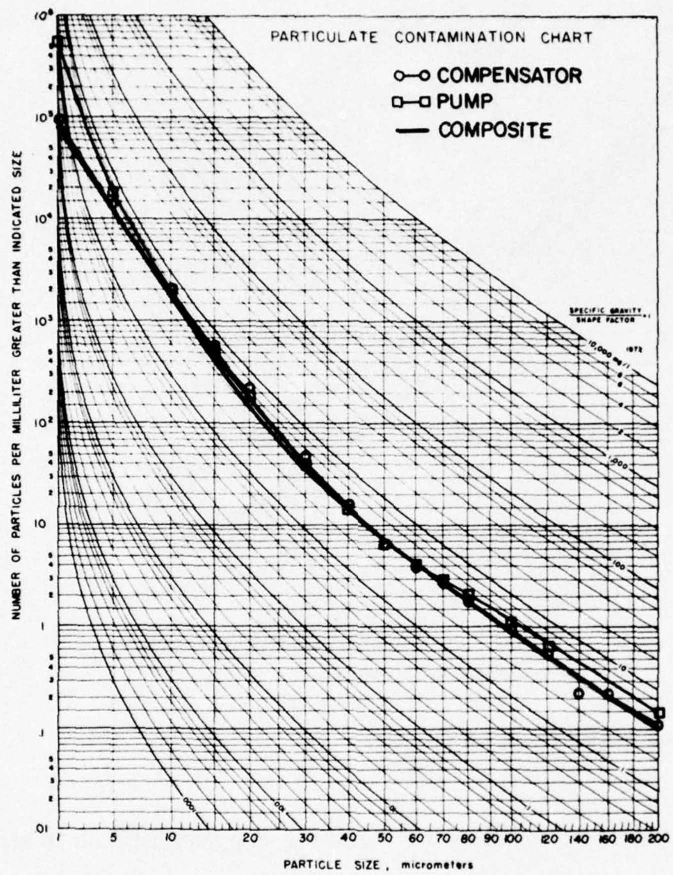


Fig. 4-9. Contaminant Tolerance Profile for Pump A1.

In discussing the 1000-hour profiles, it should be noted that, in all cases, the compensator exhibited a greater sensitivity than the pumping mechanisms, although in some cases the difference is difficult to see. For example, in the case of Pumps B2 and B3, which were tested at different system volumes, the profile of the pump is considerably higher than that of the

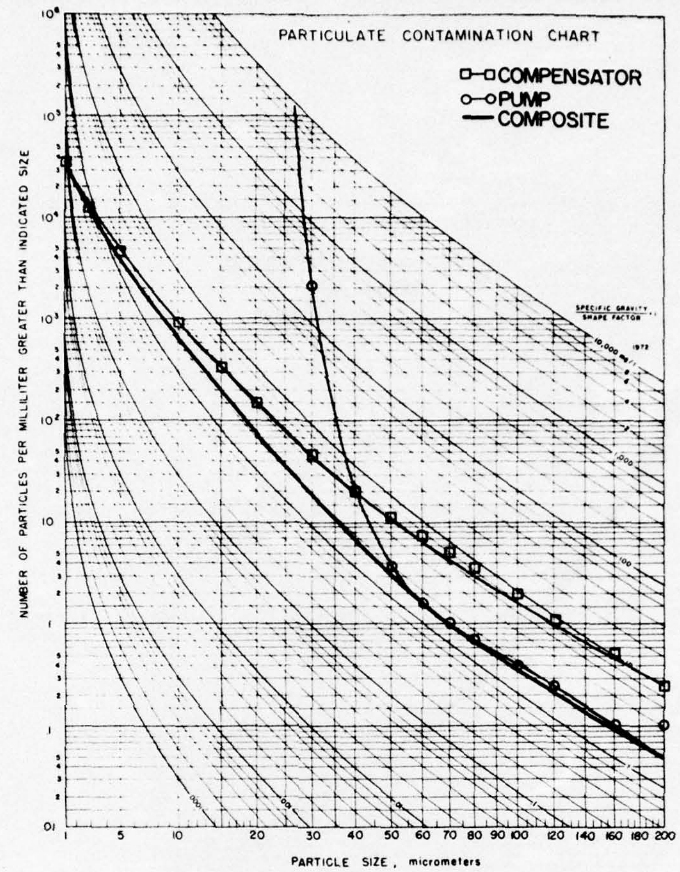


Fig. 4-10. Contaminant Tolerance Profile for Pump B2.

compensator. On the other hand, in the case of Pump C2, it is very difficult to separate the sensitivity of the two mechanisms from the data observed. Since the interpretation as provided by the profiles is somewhat subjective in nature, it is recommended that the flow and pressure ratio curves, as shown in Figs. 4-6 and 4-8, be used for specification purposes.

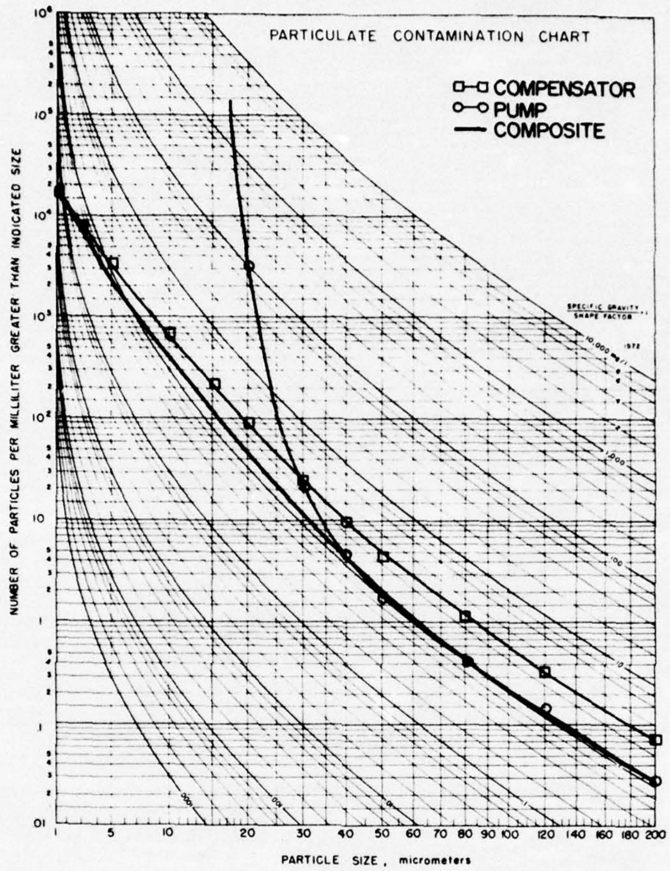


Fig. 4-11. Contaminant Tolerance Profile for Pump B3.

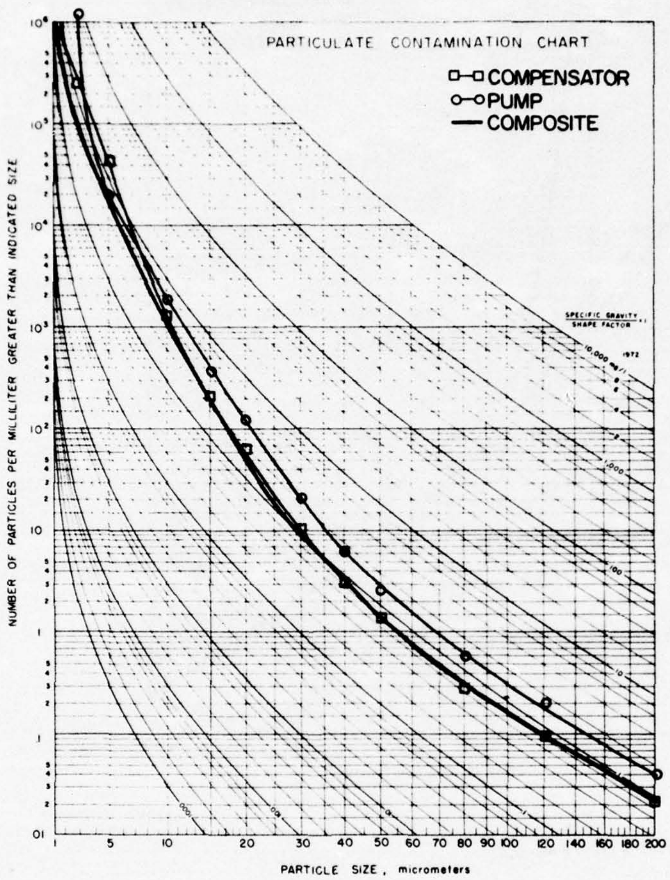


Fig. 4-12. Contaminant Tolerance Profile for Pump C2.

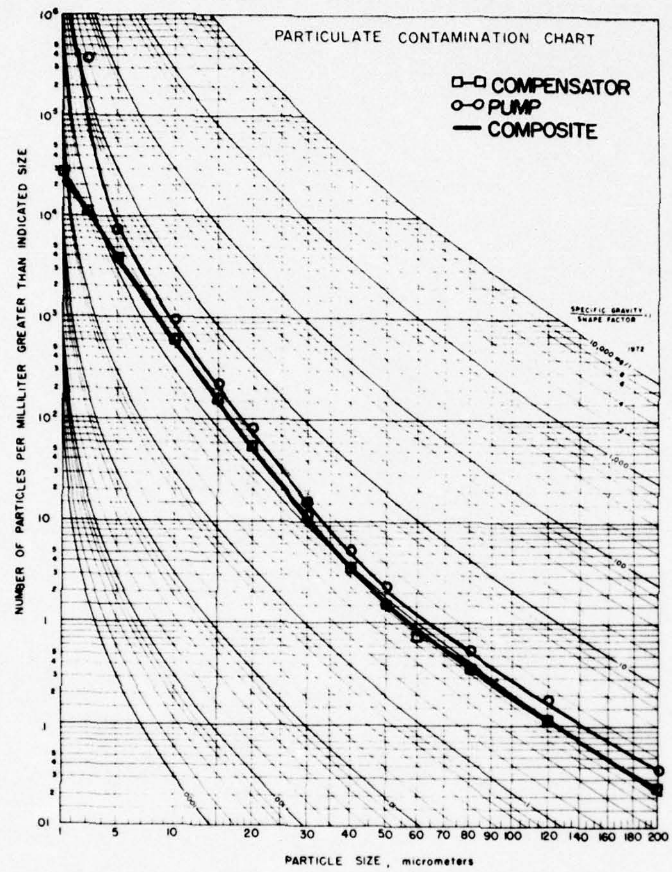


Fig. 4-13. Contaminant Tolerance Profile for Pump D.

CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

The results of this effort have shown that a single contaminant sensitivity test for a pressure compensated piston pump is indeed practical. This is an important conclusion, since previous efforts to test this type of pump have included a two-part test where the pumping mechanism was tested first and then the pump and compensator were tested together. The two-part test requires at least twice as much testing time and does not provide an appreciable amount more information. In addition, when used in a system, the pump and compensator are a single entity, and their performance as such is of greatest concern.

Another important fact which came out of this effort is associated with the manner in which the pump responds when faced with a compensator malfunction. From a system standpoint, it is more desirable for the pump outlet flow to go to zero when the compensator jams. However, as can be seen from the data presented, this is not the case with all pump designs. A closed-center system when equipped with a pump which exhibits the characteristics of a fixed-displacement type upon compensator malfunction must also be equipped with a relief valve to prevent overpressurization.

It is felt that the data obtained to date have demonstrated the repeatability of the test procedure drafted. In addition, the results from the eight tests have shown that the contaminant sensitivity test will certainly discriminate between various designs of pressure compensated piston pumps. No tests were conducted on the other seven test procedures which are needed to fully evaluate such pumps. However, many of them are basically the same as those used on fixed displacement pumps, and others are being drafted by recognized standards-making bodies. It is recommended, however, that personnel from the MERDC-OSU team continue to be a part of the effort to draft the steady-state and dynamic performance tests being pursued by NFPA.

It is unfortunate that the economic conditions of the fluid power industry did not permit a greater participation. While many companies expressed a desire to actively contribute and some did participate by donating pumps to the program, the overall response was somewhat disappointing. It is felt that, once these procedures are introduced into the activities of the cognizant standards organizations, the time will be right for a full-fledged industrial program, which is necessary to gain the desired commercial support. However, the test procedures developed and the test data acquired should certainly provide the U.S. Army with the basis for an effective procurement specification when it is needed.

— REFERENCES —

1. Bensch, L. E. and R. K. Tessmann, *"Contaminant Sensitivity Characteristics of Pressure-Compensated Pumps,"* Paper No. P73-CC-11, Basic Fluid Power Research Program, Annual Report No. 7, Fluid Power Research Center, Oklahoma State University, Stillwater, Oklahoma, 1973.

2. Bensch, L. E., and E. C. Fitch, *"A New Theory for the Contaminant Sensitivity of Fluid Power Pumps,"* Paper No. P72-CC-6, Basic Fluid Power Research Program, Annual Report No. 6, Section 72-CC, Fluid Power Research Center, Oklahoma State University, Stillwater, Oklahoma, 1972.

3. Bensch, L. E., *"Contaminant Wear Relationships for Hydraulic Motors,"* Paper No. P75-3, Basic Fluid Power Research Program, Annual Report No. 9, Fluid Power Research Center, Oklahoma State University, Stillwater, Oklahoma, 1975.

APPENDIX A

RECOMMENDED PROCEDURES FOR TESTING PRESSURE COMPENSATED HYDRAULIC FLUID POWER PUMPS

OSU-PC-1
Draft No. 1
16 February 1976

METHOD FOR EVALUATING THE STRUCTURAL INTEGRITY OF A PRESSURE COMPENSATED HYDRAULIC FLUID POWER PUMP

1. **SCOPE** To provide a method for determining the structural integrity of a pressure compensated hydraulic fluid power pump.

2. **PURPOSE** To verify the ability of a fluid power pump to maintain its structural integrity when subjected to a specified discharge pressure, shaft speed, and test duration.

3. DEFINITION

3.1 *Structural Integrity* The physical condition of a component after exposure to an above normal operating condition.

4. EQUIPMENT

4.1 *Fluid Power Test System* (See attached figure.) The high pressure, low flow auxiliary pump is utilized to supply pressure at the main pump outlet in excess of the rated pressure.

4.2 *Hydraulic Fluid* Per manufacturer's recommendation.

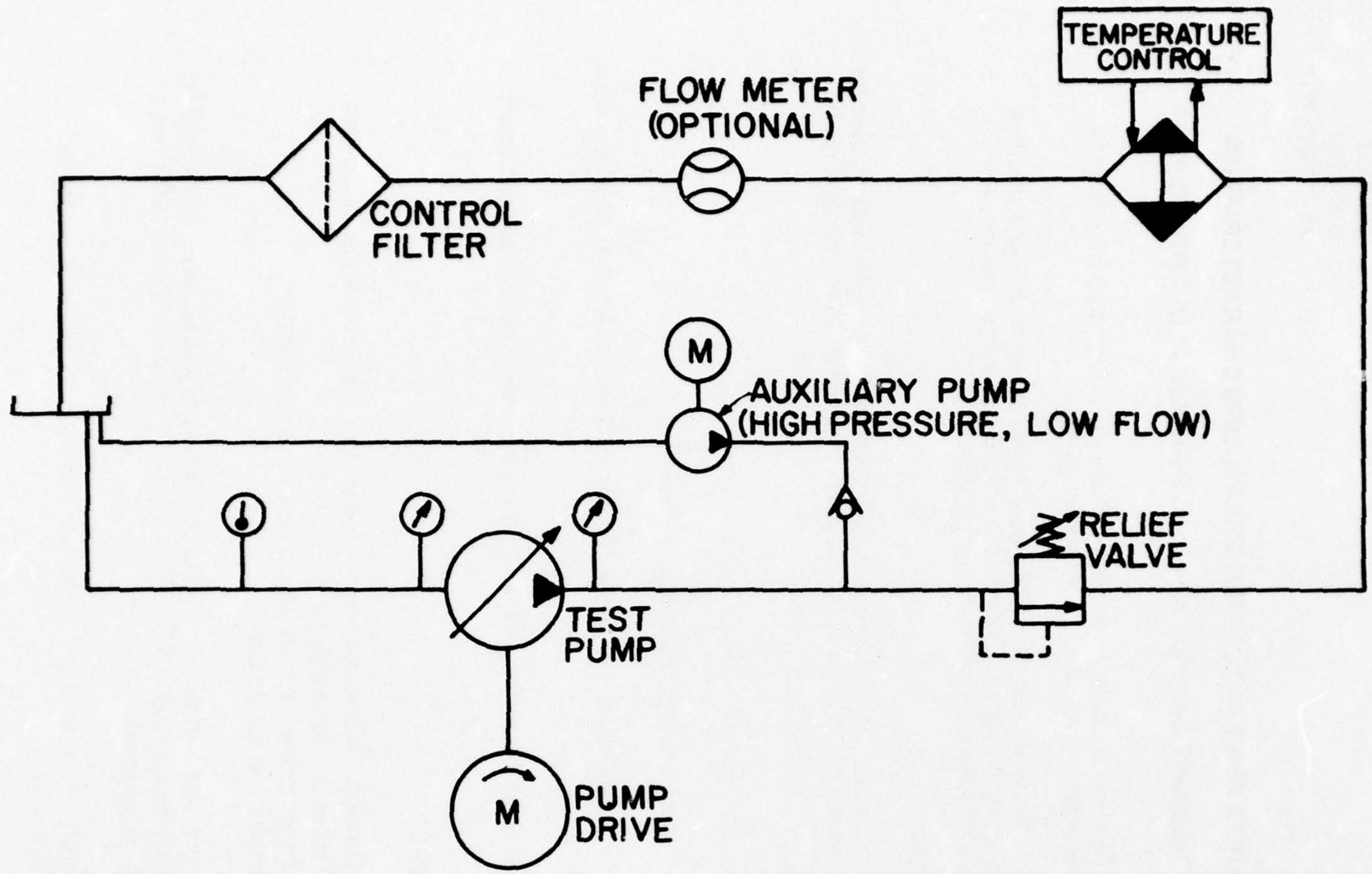
4.3 *Control Filter* Shall limit the contamination level in the system fluid to less than 10 mg/litre.

4.4 Pressure, temperature, and shaft speed measurements shall be accurate within 2%.

5. PROCEDURE

5.1 Install the pump in the test system and operate under the following conditions for not less than 60 seconds or more than 70 seconds:
(a) Speed: Manufacturer's Maximum Rated Speed
(b) Inlet Oil Temperature: 50°C
(c) Inlet Pressure: Atmospheric $\pm$ 25 mm Hg
(d) Discharge Pressure: 130% of Rated Standby or Zero Flow Pressure (Obtain 130% pressure by utilizing the auxiliary pump and relief valve.)

5.2 Record any evidence of external leakage, including formation of drops or wetting of external surfaces.



Fluid Power Test System, OSU-PC-1

OSU-PC-2
Draft No. 1
16 February 1976

METHOD FOR EVALUATING THE FILLING CHARACTERISTICS OF A PRESSURE COMPENSATED, HYDRAULIC FLUID POWER PUMP

1. **SCOPE** To provide a method for determining the filling characteristics of a pressure compensated, hydraulic fluid power pump.

2. **PURPOSE** To verify the ability of a fluid power pump to deliver a specified flow rate when operating at rated speed, a reduced discharge pressure, and specified inlet pressure, temperature, and system fluid.

3. **DEFINITION**

3.1 *Filling Characteristic* A feature exhibited by a fluid power pump which indicates the void volume within the pumping chambers at specified operating conditions.

4. **EQUIPMENT**

4.1 *Fluid Power Test System* (See Fig. 1.)

4.2 *Fluid* Per Manufacturer's Recommendation

4.3 *Control Filter* Shall limit the contamination level in the system fluid to less than 10 mg/litre.

4.4 Pressure, temperature, shaft speed, and flow rate measurements shall be accurate within 2%.

5. **PROCEDURE**

5.1 Install the pump in the test system and operate under the following conditions:
(a) Inlet Pressure: Atmospheric $\pm$ 25 mm Hg
(b) Inlet Temperature: Per Manufacturer's Recommendation
(c) Discharge Pressure: 35 bar

5.2 Vary the pump speed from 600 RPM to manufacturer's rated speed and record the flow rate shaft speeds of 600 RPM, 50% rated, 75% rated, 85% rated, 90% rated, and at 100% rated speed.

5.3 At rated speed, reduce the inlet pressure to atmospheric minus 25 cm Hg and record the flow rate.

5.4 Plot flow rate versus pump speed and indicate the flow rate at the reduced inlet pressure, as shown in Fig. 2.

6. INTERPRETATION

6.1 Using the test data, complete Table I.

6.2 Calculate the best fit (least squares) flow rate (Q_B) for each speed value using the following equation:

$$Q_B = aN + b$$

where:

$$a = \frac{7C - BA}{7D - A^2}$$

$$b = \frac{B - aA}{7}$$

$Q_x \triangleq Q_B$ for N = Rated Speed

$Q_y \triangleq Q$ for N = Rated Speed, and

Inlet Pressure = Atmospheric – 25.4 cm Hg

TABLE I

Speed N (RPM)	Flow Rate Q (LPM)	N x Q	N^2	Best Fit Q_B	ΔQ $\lvert Q - Q_B \rvert$	.03Q
1						
2						
3						
4						
5						
6						
7						
Σ= A	Σ= B	Σ= C	Σ= D			

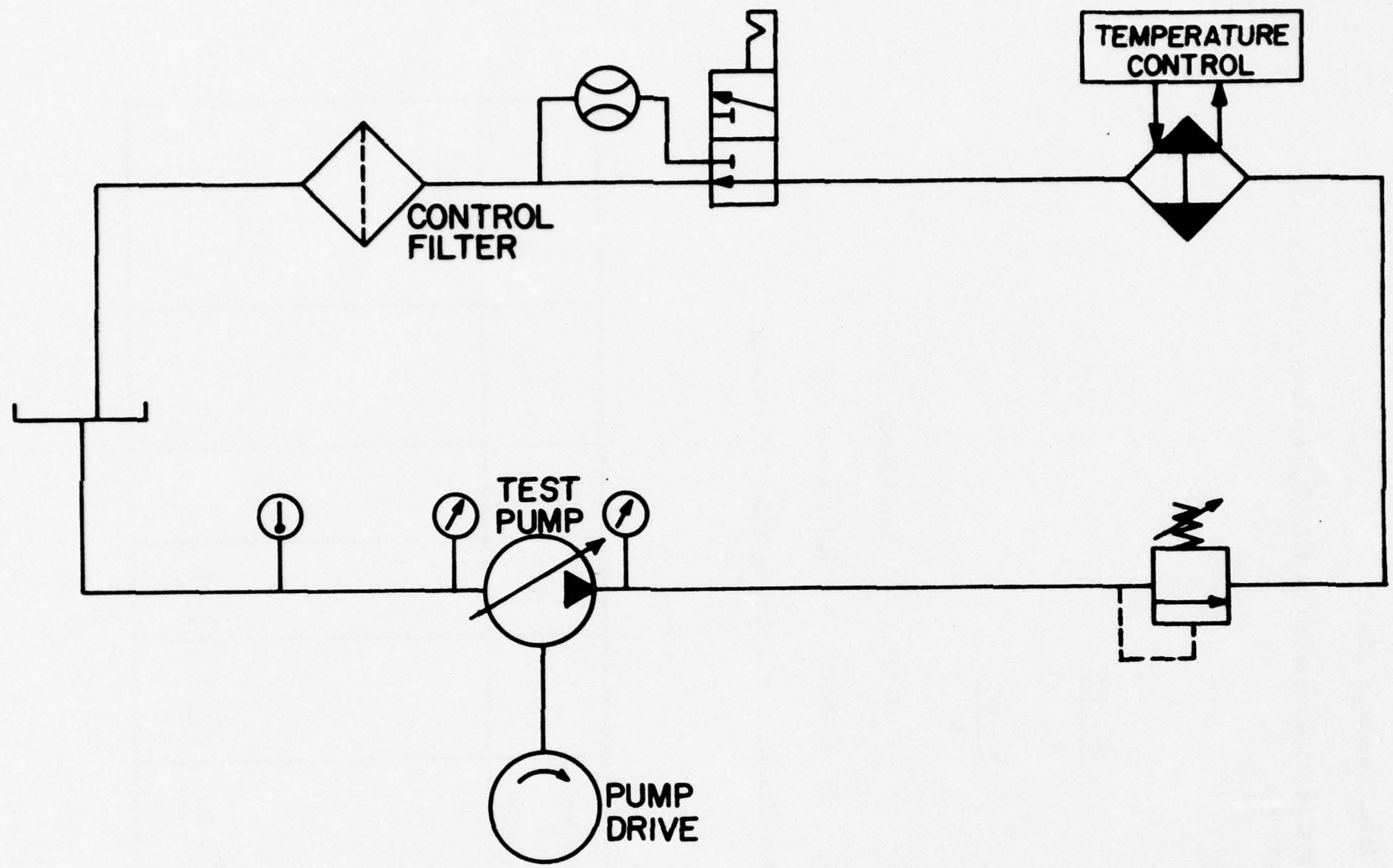


Fig. 1. Fluid Power Test System for OSU-PC-2.

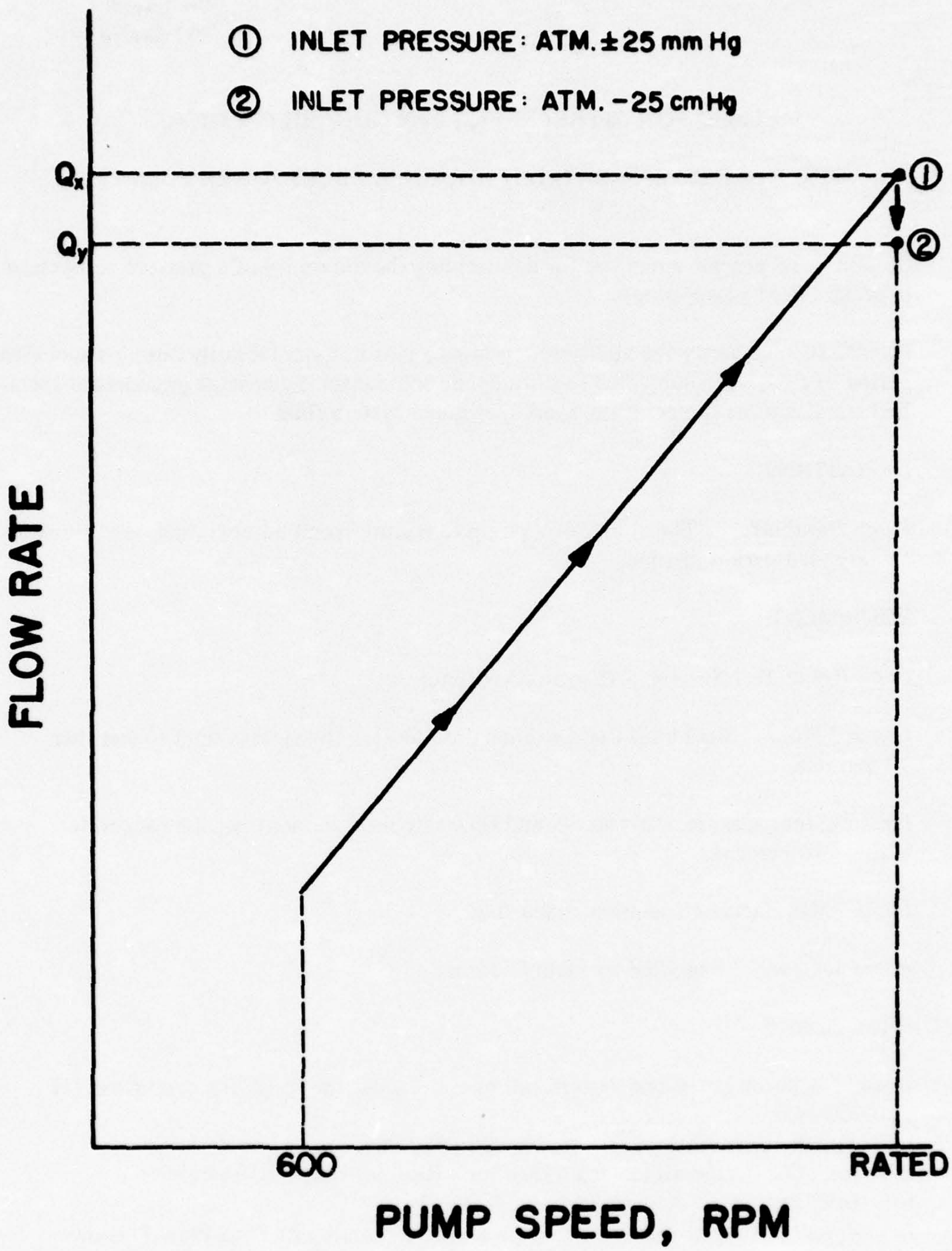


Fig. 2. Flow Rate Versus Pump Speed, OSU-PC-2.

OSU-PC-4
Draft No. 1
16 February 1976

METHOD FOR ESTABLISHING THE DURABILITY OF A PRESSURE COMPENSATED HYDRAULIC FLUID POWER PUMP

1. **SCOPE** To provide a method for determining the durability of a pressure compensated hydraulic fluid power pump.

2. **PURPOSE** To verify the ability of a pump to perform satisfactorily during a specified period of time when subjected to both cyclic and constant discharge pressures at specified conditions of temperature, shaft speed, and system fluid.

3. **DEFINITION**

3.1 *Pump Durability* The ability of a pump to endure specified operating conditions for an extended period of time.

4. **EQUIPMENT**

4.1 *Fluid Power Test System* (See attached figure.)

4.2 *Control Filter* Shall limit the contamination level in the system fluid to less than 10 mg/litre.

4.3 Pressure, temperature, shaft speed, and flow rate measurements shall be accurate within two percent.

4.4 *Fluid* Manufacturer's recommended fluid.

4.5 *Rated Pressure* Specified by manufacturer.

5. **PROCEDURE**

5.1 Install the pump in the test system and operate under the following conditions for 500,000 cycles:
(a) Speed: Manufacturer's Rated Speed +0%/-20%
(b) Inlet Oil Temperature: Manufacturer's Recommended Temperature
(c) Inlet Pressure: Atmospheric $\pm$ 25 mm Hg.
(d) Pressure Range: From 5 to 100% of Rated Standby or Zero Flow Pressure (Transient pressure peak shall not exceed 130% of rated standby pressure.)
(e) Endurance Cycle Conditions: 60 Cycles/Minute (one-half second at a pressure of at least 100% of rated and 0.2 seconds at 5% of rated pressure – rate of pressure rise shall not exceed 11 kbar/second).

5.2 Subject the pump to a constant pressure test under the following conditions:

(a) Speed: Manufacturer's Rated Speed +0%/-20%

(b) Inlet Oil Temperature: Manufacturer's Recommended Temperature

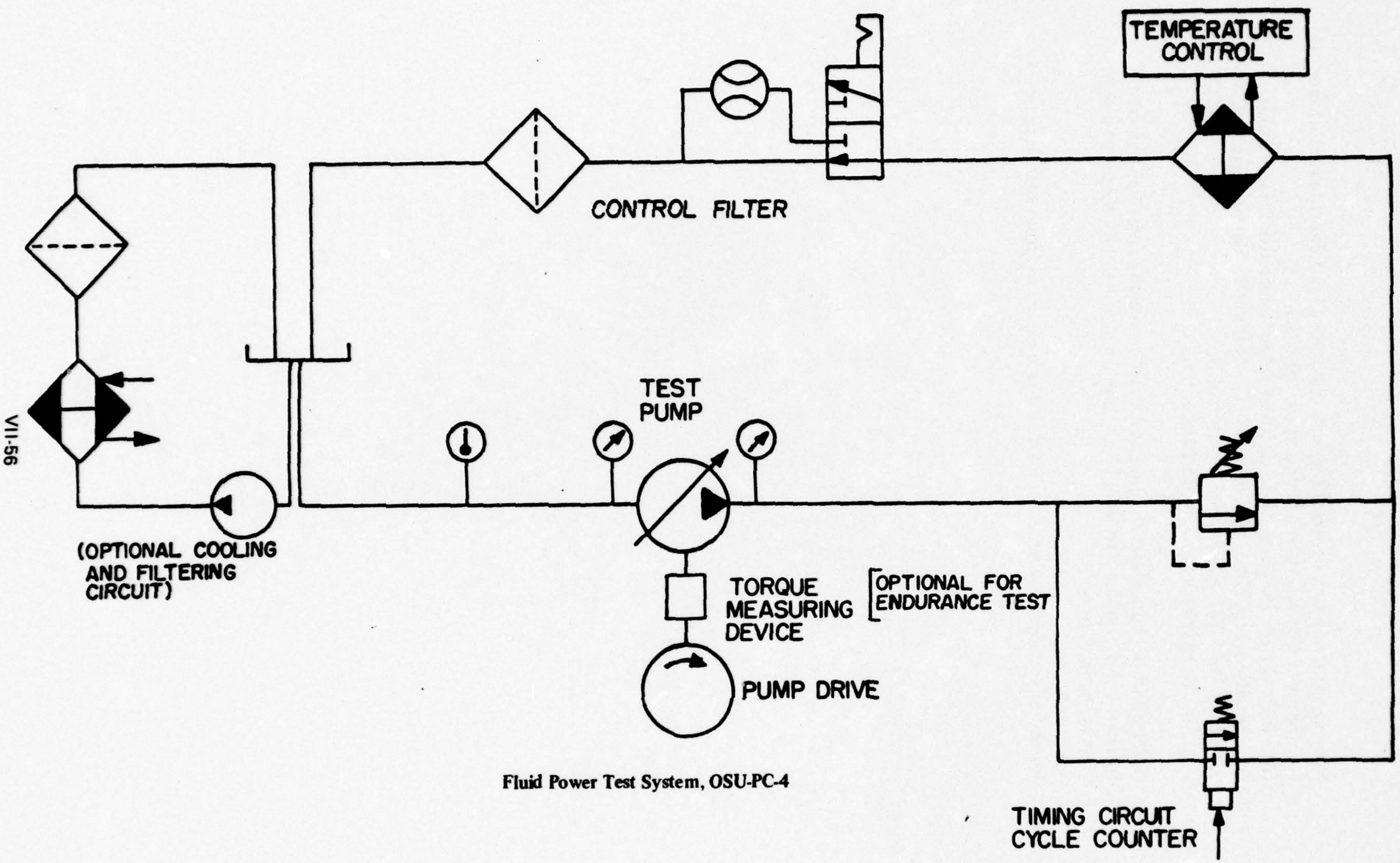
(c) Duration Test Time = 50 Hours / (1 – 100 x % Deviation from Rated Speed)

(d) Inlet Pressure: Atmospheric ± 25 mm Hg

(e) Discharge Pressure: 75% Rated, Continuous

(Inlet oil throughout the endurance test must be visually free of entrained air. Inspection for entrained air shall be accomplished visually by means of a sight glass in the inlet line.)

5.3 Record any evidence of external leakage and measure external shaft leakage.



Fluid Power Test System, OSU-PC-4

OSU-PC-5
Draft No. 1
16 February 1976

METHOD FOR EVALUATING THE LOW SPEED AND HIGH SPEED PERFORMANCE OF A PRESSURE COMPENSATED, HYDRAULIC FLUID POWER PUMP

1. **SCOPE** *To provide a method for determining the ability of a pressure compensated,* hydraulic fluid power pump to survive under low and high speed operation.

2. **PURPOSE** To verify the ability of a pump to withstand both overspeed and underspeed operation and perform satisfactorily at specified conditions of temperature, discharge pressure, and system fluid.

3. **EQUIPMENT**

3.1 *Fluid Power Test System* (See attached figure.)

3.2 *Fluid* Per manufacturer's recommendation.

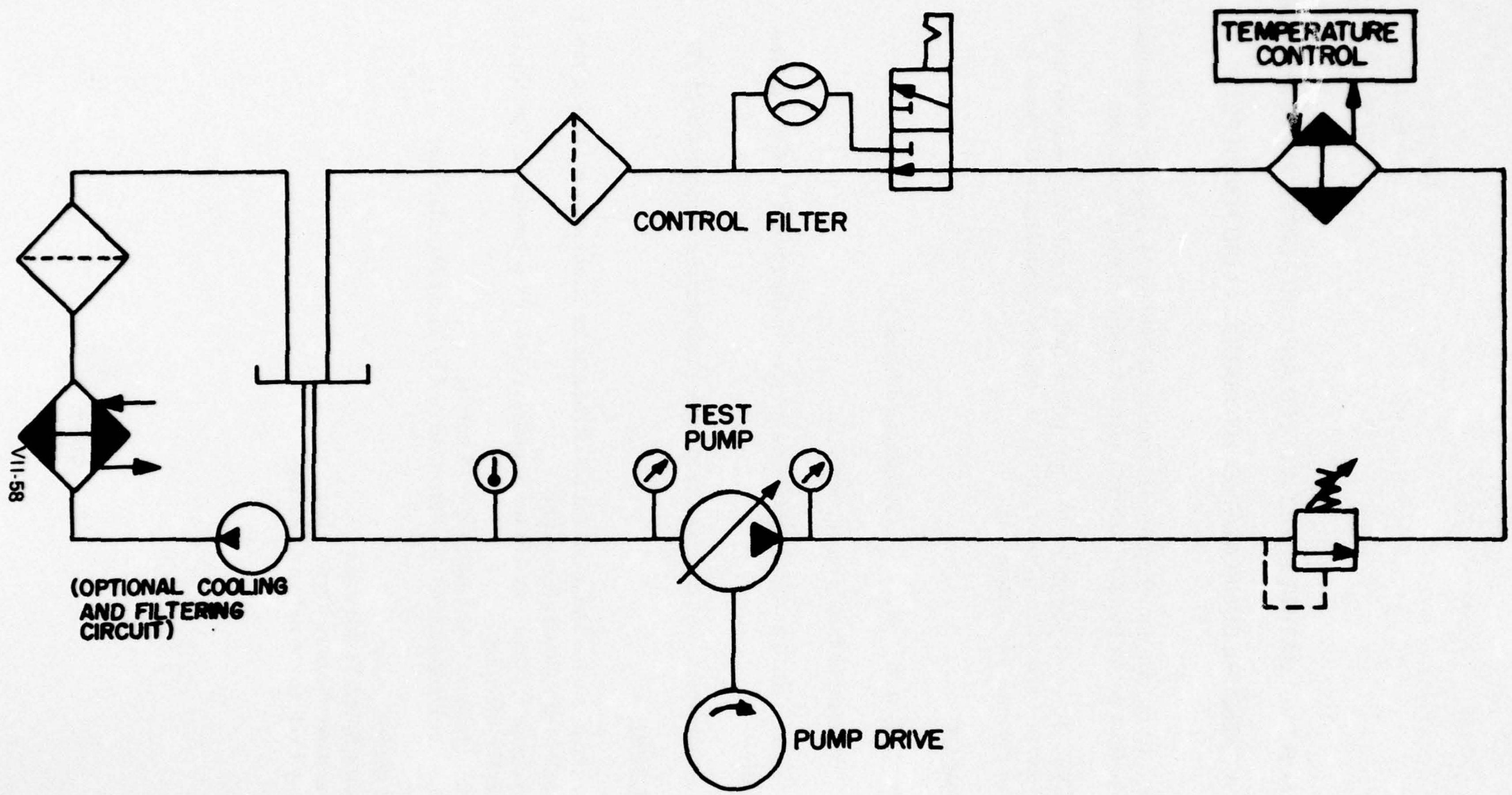
3.3 *Control Filter* Shall limit the contamination level in the system fluid to less than 10 mg/litre.

3.4 Pressure, shaft speed, and temperature measurements shall be accurate within 2%.

4. **PROCEDURE**

4.1 Operate the pump for one minute at 600 RPM and for 30 minutes at 110% of rated speed under the following conditions:
(a) Discharge Pressure: 75% Rated Standby or Zero Flow Pressure as Specified by the Manufacturer
(b) Inlet Pressure: *Atmospheric ± 25 mm Hg*
(c) Inlet Oil Temperature: As Recommended by Pump Manufacturer

4.2 Record the following:
(a) Pump Speed Versus Time
(b) Discharge Pressure Versus Time
(c) Any Evidence of External Leakage



Fluid Power Test System for OSU-PC-5.

OSU-PC-6
Draft No. 1
16 February 1976

METHOD FOR EVALUATING THE LOW TEMPERATURE PERFORMANCE OF A PRESSURE COMPENSATED, HYDRAULIC FLUID POWER PUMP

1. **SCOPE** To provide a method for determining the ability of a pressure compensated, hydraulic fluid power pump to survive under low temperature conditions.

2. **PURPOSE** To verify the ability of a pump to withstand low temperature operation and perform satisfactorily at specified conditions of speed, discharge pressure, and system fluid.

3. EQUIPMENT

3.1 Cold room capable of maintaining an air temperature of -30°C.

3.2 Fluid power test circuit as shown in Fig. 1.

3.3 Control filter shall limit the contamination level in the system fluid to less than 10 mg/litre.

3.4 Pressure, shaft speed, and temperature measurements shall be accurate within 2%.

4. PROCEDURE

4.1 Install the pump in the test system.

4.2 Adjust the total system volume (litres) to be numerically equal to one-half the pump delivery rate (litres/minute) at maximum steady-state speed.

4.3 Use manufacturer's recommended fluid.

4.4 Circulate system fluid through *"clean-up"* filter for 15 minutes.

4.5 Bypass *"clean-up"* filter.

4.6 Lower the temperature of the pump and hydraulic system to -30°C and maintain this temperature for a period of at least 12 hours.

4.7 Operate the pump in accordance with the temperature schedule given in Fig. 2. The inlet pressure of the pump shall not be less than the pressure specified by the manufacturer. Note that the test parameter which dictates changes in speed and pressure is inlet temperature (T_i).

4.8 Terminate the test when the inlet temperature reaches -15°C.

5. PRESENTATION OF RESULTS

5.1 Record pressure versus time.

5.2 Record temperature versus time.

5.3 Record pump speed versus time.

5.4 Record total shaft seal leakage in millilitres.

5.5 Report any evidence of external leakage occuring other than shaft seal leakage.

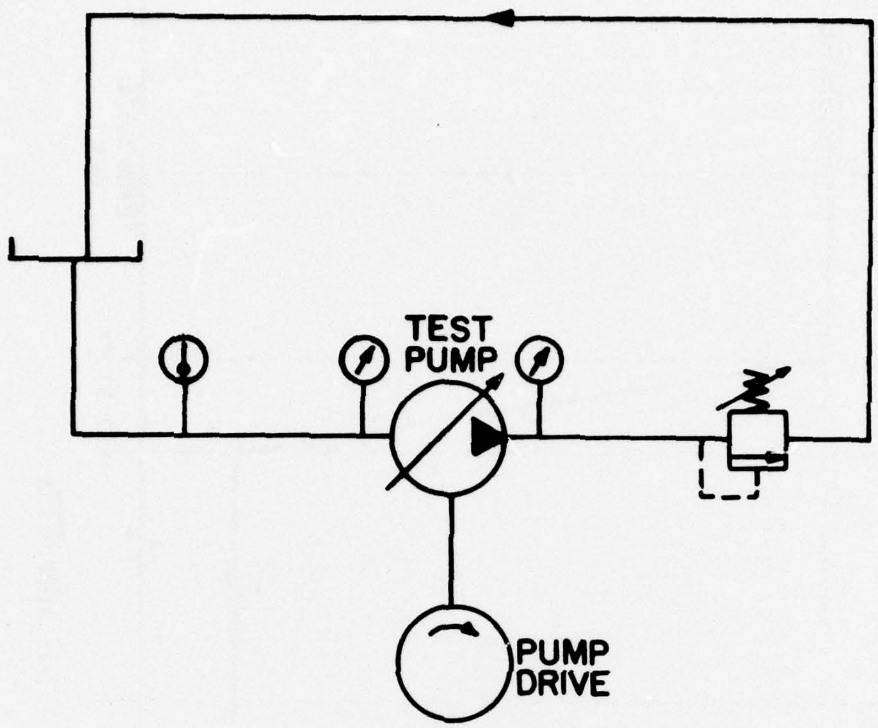


Fig. 1. Fluid Power Test System, OSU-PC-6.

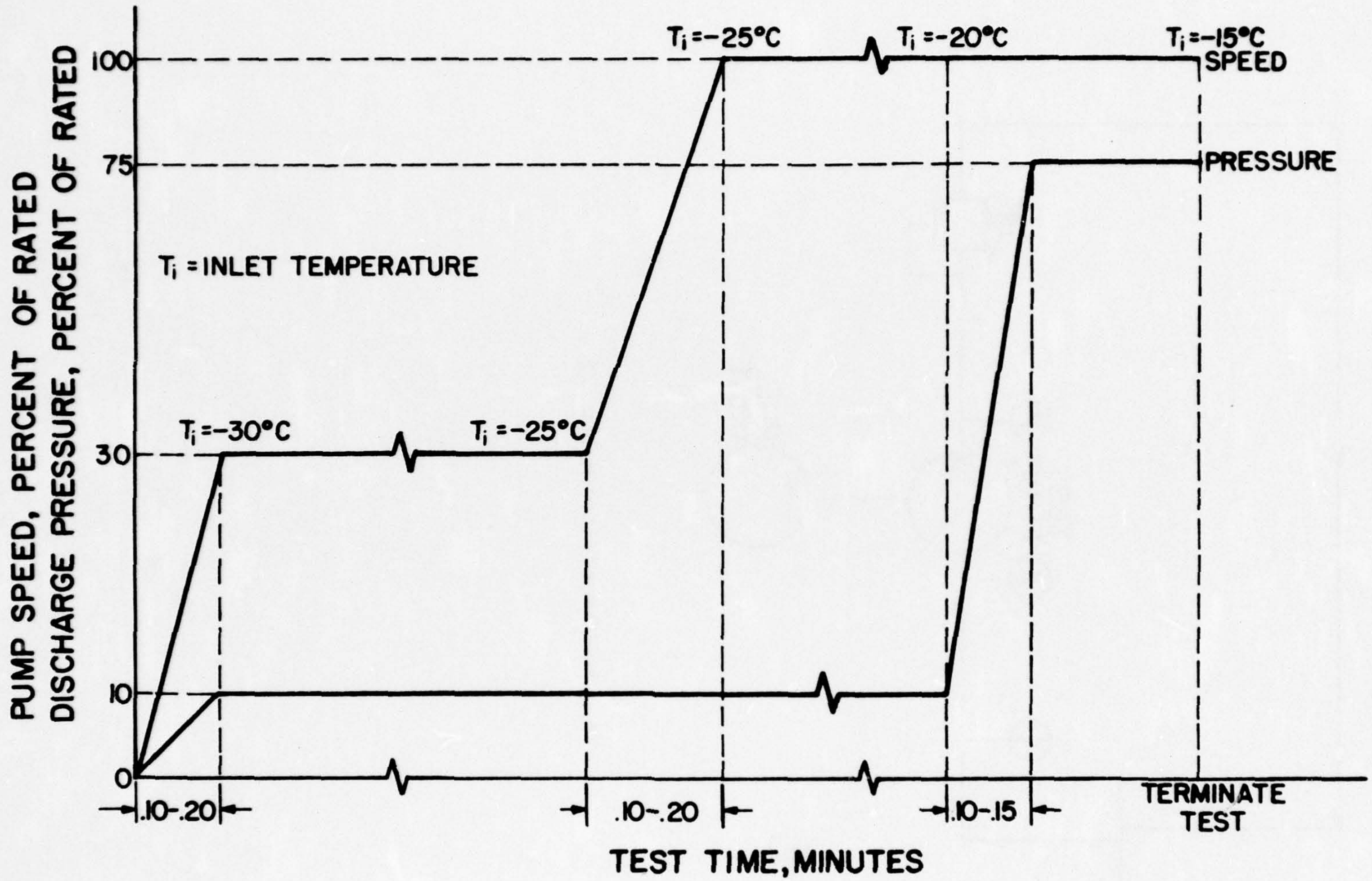


Fig. 2. Pump Speed, Pressure Schedule, OSU-PC-6.

OSU-PC-7
Draft No. 2
16 February 1976

METHOD OF ESTABLISHING THE CONTAMINANT SENSITIVITY OF A PRESSURE COMPENSATED HYDRAULIC FLUID POWER PUMP

1. INTRODUCTION

1.1 *General* The useful life of a hydraulic pump is directly related to its assembled configuration, fabrication materials, hydraulic fluid, and its operating conditions.

The life of a pressure compensated pump is considered to be terminated when it can no longer deliver a specified flow rate at a given shaft speed, discharge pressure, and fluid condition or when the standby pressure decreases below a specified level.

A pump may reach a terminal state due to catastrophic (mechanical interference or material overstress) failure or by the cumulative effect of wear processes.

The rate of wear within hydraulic pumps is proportional to the contamination level of the hydraulic fluid exposed to the internal surfaces of the pump.

Any wearing away of the surfaces which form the critical clearance spaces (leakage paths) of a hydraulic pump will be accompanied by a measurable degradation in its delivered flow rate.

Any critical wear associated with the compensating mechanism can be measured by a change in the standby or zero flow pressure.

The fabrication materials together with the characteristic size and shape of critical clearance spaces within a pump uniquely establish a contaminant sensitivity for a given operating condition.

Based upon the above considerations, pressure and flow degradation ratios are plotted for comparison of the contaminant sensitivity of hydraulic pumps at the same reference operating conditions.

1.2 *Scope* This method establishes a procedure for evaluating the contaminant sensitivity of a hydraulic pressure compensated pump to abrasive contaminant.

The flow rate and standby pressure at specified operating conditions are used as the performance parameters. Pump failure by processes other than surface wear is not considered in this procedure.

1.3 *Purpose* It is the purpose of this method to provide a reliable, economical, and accelerated test procedure for appraising the contaminant sensitivity of hydraulic pressure compensated pumps.

1.4 *Outline of Method* Flow rate and standby pressure are recorded as the pump is operated at rated conditions after being subjected to specific contaminant size ranges. A constant gravimetric level of 300 mg/litre is used for all particle size ranges. The size ranges used in this procedure are from zero to 5, 10, 20, 30, 40, 50, 60, 70, and 80 micrometres. Based upon the test results, degradation characteristics of the pump can be established.

2. DEFINITIONS

2.1 *Contaminant Sensitivity* The sensitivity of a hydraulic component to the presence of contaminant.

2.2 *Flow Degradation Ratio* The ratio of the flow after contaminant exposure to the initial measured flow (Q_r).

2.3 *Pressure Degradation Ratio* The ratio of the standby pressure after contaminant exposure to the initial measured standby pressure (P_r).

3. EQUIPMENT AND SUPPLIES

3.1 Hydraulic test circuit as illustrated in Fig. 1.

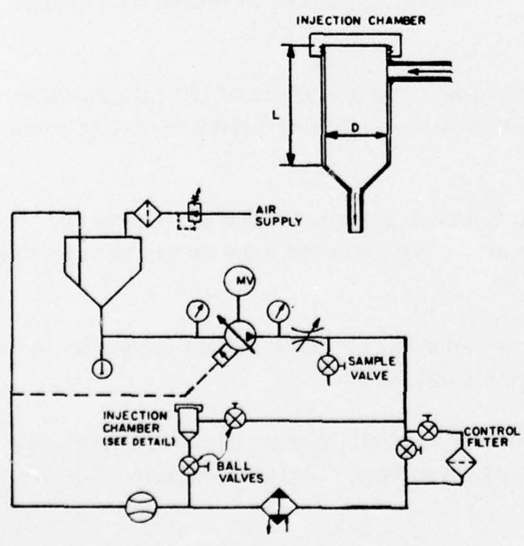


Fig. 1. Hydraulic Test Circuit Schematic.

3.2 Facility for measuring the gravimetric level of a fluid.

3.3 Supply of classified AC Fine Test Dust.

3.4 Supply of clean fluid sample bottles with an RCL below 10 particles per millilitre.

3.5 Supply of clean slurry injection bottles.

3.6 Test fluid recommended by pump manufacturer.

4. TEST FACILITY REQUIREMENTS

4.1 The test system illustrated in Fig. 1 shall be comprised of a reservoir, injection system heat exchanger, flowmeter, pressure gages, temperature indicator, control filter, test pump, pump drive, hydraulic fluid, and load valve.

4.2 The reservoir shall be constructed with a conical bottom displaying a projected angle of not more than 90°. The oil entering the reservoir shall be diffused below the surface of the oil. Provisions should be made to pressure the reservoir.

4.3 The injection chamber shall be constructed as shown in Fig. 1. The volume of the chamber should be approximately 500 ml, and the ratio L/D shall be 10. The included angle at the bottom of the chamber shall be less than 90°.

4.4 The heat exchanger shall be either a one- or two-pass unit and shall be mounted vertically with the oil entering from the bottom. Oil shall be circulated through the tube side and the water through the shell side.

4.5 The flowmeter should be insensitive to contaminant.

4.6 Flow, pressure, temperature, and speed measurements shall be accurate within two percent of the desired values.

4.7 The control filter shall be capable of providing a contaminant background of less than 10 mg/litre.

4.8 The test pump shall be operated at manufacturer's recommended speed, inlet pressure, and inlet temperature.

4.9 The test system shall not exhibit the presence of entrained air.

4.10 The lines connecting the hydraulic components shall be sized such that turbulent mixing exists throughout. Precaution shall be taken to insure against contaminant traps, silting areas, and combinations of cyclonic separation zones and quiescent chambers.

5. SYSTEM VERIFICATION

5.1 Install a pump which is known to be relatively insensitive to contamination in the test circuit.

5.2 Adjust the system volume (exclusive of the filter system) so that it is numerically equal to one-fourth (± 10%) of the lowest volume flow rate (per minute) to be used for testing.

5.3 Circulate through the filtering system until the contaminant background is less than 10 mg/litre.

5.4 Bypass the filter system.

5.5 Add a quantity of AC Fine Test Dust to the system to bring the contamination level to 300 mg/litre. The contaminant should be injected in the form of a well-mixed slurry to prevent agglomeration of the particles.

5.6 Operate the system at the minimum flow rate to be used for testing.

5.7 Extract a fluid sample at 15-minute intervals from the system.

5.8 Repeat 5.6 and 5.7 until four samples are obtained. The system should be run continuously during this period.

5.9 Measure the gravimetric level of each sample.

5.10 Circulate through the filter until the contaminant background is less than 10 mg/litre.

5.11 Compute the average of the four gravimetric levels obtained in 5.9. The system is qualified for testing if none of the levels differs from the average level by more than 10% and the average level is 300 mg/litre ± 15 mg/litre.

6. PRELIMINARY PREPARATION

6.1 Install the test pump in the circuit.

6.2 Operate the pump at rated conditions of speed and temperature with the filter system in the circuit. Use the following schedule for pump operating pressure: 15 minutes at 25% standby pressure, 15 minutes at 50% standby pressure, and 15 minutes at 75% standby pressure.

6.3 Measure the flow rate at 67% standby pressure.

6.4 Increase the pressure until the flow equals one-half the flow rate at 67% standby pressure and operate for 60 minutes.

6.5 Block the outlet of pump and record initial standby pressure (P_r).

6.6 Reduce the pressure to 67% of P_r and record the initial flow (Q_r).

6.7 Adjust the system volume (exclusive of the filter system) so that it is numerically equal to one-fourth ($\pm$ 10%) of the initial flow rate (per minute)(Q_r).

7. CONTAMINANT SENSITIVITY TEST

7.1 Determine the quantity of contaminant (g_i) required for each injection according to the following expression: g_i (grams) = 0.3 x (Volume of System in Litres). NOTE: The *gravimetric level of the system will be 300 mg/litre for each size range.*

7.2 Prepare a slurry containing g_i grams of contaminant of each of the following size ranges: 0-5, 0-10, 0-20, 0-30, 0-40, 0-50, 0-60, 0-70, and 0-80 micrometres.

7.3 Operate the test pump at rated conditions.

7.4 Adjust pressure to achieve a flow rate of 0.5 $Q_r \pm$ 10%.

7.5 Bypass the filter system.

7.6 Inject the 0-5 micrometre slurry.

7.7 Continue operating at the specified flow rate for 30 minutes.

7.8 Reduce the pressure to 0.67 $P_r \pm$ 1%.

7.9 Circulate through the filter system until the contaminant background is less than 10 mg/litre.

7.10 Record the final flow rate with the pump operating as in 7.8.

7.11 Block outlet of pump and record the final standby pressure.

7.12 If the final flow rate of Step 7.10 has decreased to less than 70% of the initial value Q_r or the final standby pressure of Step 7.11 has changed by more than $\pm$ 30% from the initial standby pressure P_r, proceed directly to Step 8.

7.13 Repeat Steps 7.4 through 7.12 using 0-10, 0-20, 0-30, 0-40, 0-50, 0-60, 0-70, 0-80 micrometre contaminant in progressively increasing sizes.

8. *PRESENTATION OF TEST RESULTS*

8.1 Record pump identification, operating conditions, and test data as shown in Table 1.

Table 1: Contaminant Sensitivity Test Results.

Date Tested: ______	**Test Location:** ______
Pump: ______	**Grams Injected:** ______
Speed: ______	Q_r = ______
Temperature: ______	P_r = ______
Fluid: ______	**Comments:** ______
System Volume: ______	______

Size Injected (μM	*Final Flow Rate (7.10)*	*Flow Degradation Ratio (8.2)*	*Final Standby Pressure (7.11)*	*Pressure Degradation Ratio (8.3)*
0-5				
0-10				
0-20				
0-30				
0-40				
0-50				
0-60				
0-70				
0-80				

8.2 Calculate the flow degradation ratio for each injection by dividing the final flow rate from 7.10 by the initial flow rate from 6.6. Record in Table 1 to three decimal places.

8.3 Calculate the pressure degradation ratio for each injection by dividing the final standby pressure from 7.11 by the initial standby pressure from 6.5. Record in Table 1 to 3 decimal places.

8.4 Plot the flow degradation ratios calculated in 8.2 on linear coordinates versus the respective maximum particle size for each injection.

8.5 Plot the pressure degradation ratios calculated in 8.3 on linear coordinates versus the respective maximum particle size for each injection.

TEST VERIFICATION DRAFT

16 February 1976

NOTE: This draft was prepared by personnel at Oklahoma State University as part of a program with the U.S. Army Mobility Equipment Research and Development Center.

(PROPOSED)

COMPONENT PERFORMANCE SPECIFICATION

PUMPS, HYDRAULIC FLUID, PRESSURE COMPENSATED, PISTON

210 BAR SERVICE

1.0 SCOPE

1.1 *Definition* A 210 bar rated, pressure compensated hydraulic pump is defined as one capable of operating adequately in a fluid system designed for variable flow operation and a maximum pressure of 210 bar.

1.2 *Inclusion* This specification includes those aspects of a 210 bar rated, pressure compensated, hydraulic fluid power pump which are concerned with its environment and operation. Maximum rated speed of the pump is established by the pump manufacturer. The system fluid conforms to Mil-L-2104B, Class 10, or Mil-L-10295.

2.0 PURPOSE

2.1 *Requirements* This specification establishes the specific requirements of the pump with regard to: (1) environmental conditions, (2) operation, and (3) test sequence.

2.2 *Test Procedures* This specification requires the use of test procedures developed by Oklahoma State University under the sponsorship of the U.S. Army MERADCOM, which are cited in each requirement.

2.3 *Test Conditions* Unless otherwise specified, the test conditions shall be:

RATED STANDBY PRESSURE: 210 Bar **FLUID: Mil–L-2104B, Class 10**
INLET PRESSURE: Atmospheric $\pm$ 25 mm Hg
INLET TEMPERATURE: 50°C
RATED SPEED: Per manufacturer's recommendation (as established by 3.2.2 of this specification)

3.0 REQUIREMENTS

3.1 *Environmental Conditions* The pump shall be constructed to operate throughout the entire range of environmental conditions specified herein.

3.1.1 *Temperature* The pump shall be subjected to the Low Temperature Test per the OSU-PC-6 Standard Test Procedure. Inability of the pump to rotate without damage, inability of the pump to develop 157.5 bar under the specified conditions or to exhibit external leakage exceeding three drops per hour shall constitute failure of the pump. The fluid for this test shall be Mil-L-10295, and the inlet pressure shall be not less than 25 cm Hg.

3.1.2 *Contamination Level* The pump shall be subjected to the Contaminant Tolerance Test per OSU-PC-7 Standard Test Procedure. The resulting flow and pressure degradation must be within the limits specified in Figs. 1 and 2 respectively.

3.2 *Operation*

3.2.1 *Proof Pressure* The pump shall be subjected to the OSU-PC-1 Standard Test Procedure. Evidence of external leakage in the form of drops from the shaft seal or wetting of external surfaces shall be cause for rejection of the pump.

3.2.2 *Filling Characteristics* The pump shall be subjected to the OSU-PC-2 Standard Test Procedure. For each test speed (N), the following relation shall hold: $\Delta Q \leq 0.03Q$. In addition, the pump shall satisfy the following relation at rated speed: $Q_y \geq 0.95 Q_x$.

3.2.3 *Steady State and Dynamic Performance* The pump shall be subjected to the OSU-PC-3 and OSU-PC-8 Standard Test Procedures. The pump shall exhibit an overall efficiency of at least 77% over the indicated pressure range. Additional requirements to be established when PC-3 and PC-8 are completed.

3.2.4 *Endurance* The pump shall be subjected to OSU-PC-4 Standard Test Procedure. Malfunction prior to completion of this test or evidence of external leakage exceeding three drops per hour shall constitute failure of the pump.

3.2.5 *Speed Performance* The pump shall be subjected to OSU-PC-5 Standard Test Procedure. Inability of the pump to develop 157.5 bar or evidence of external leakage exceeding three drops per hour shall be cause for rejection of the pump.

3.2.6 *Final Efficiency* The pump shall be subjected to OSU-PC-3 Standard Test Procedure. The pump shall exhibit an overall efficiency of at least 73% over the indicated pressure range. Additionally, the flow rate of the pump shall not have decreased more than 10% from the flow rates given by the initial efficiency test (3.2.3).

3.3 *Test Sequence* Two pumps are required for the tests covered by this specification. One pump shall be subjected to the following test sequence:

(a) Proof Pressure Test Per OSU-PC-1
(b) Filling Characteristics Test Per OSU-PC-2
(c) Contaminant Tolerance Test Per OSU-PC-7

The second pump shall be tested in the following sequence:

(a) Steady-State & Dynamic Performance Per OSU-PC-3 and OSU-PC-8
(b) Endurance Test Per OSU-PC-4
(c) Performance Test Per OSU-PC-5
(d) Low Temperature Test Per OSU-PC-6
(e) Final Efficiency Test Per OSU-PC-3

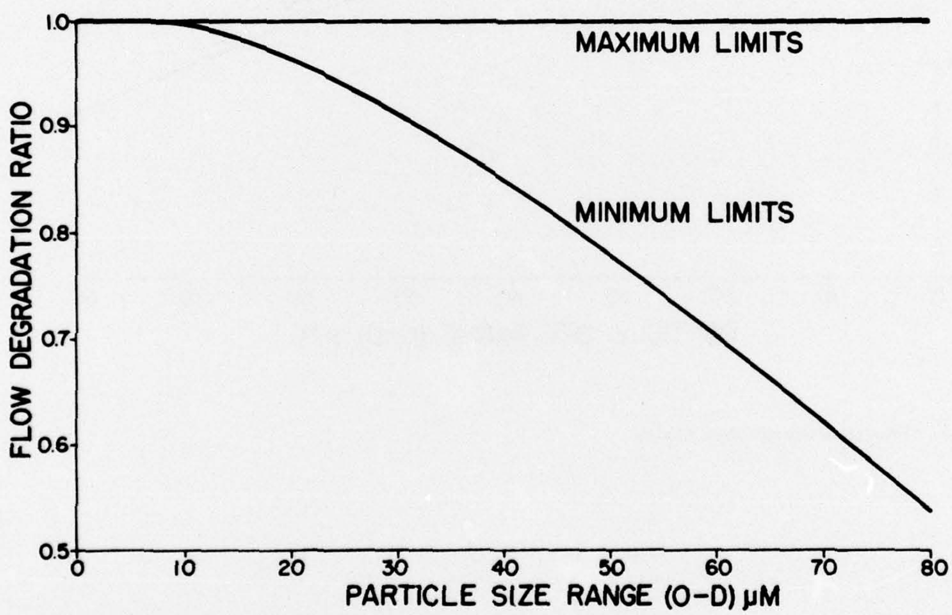


Fig. 1. Flow Degradation Specification Limits.

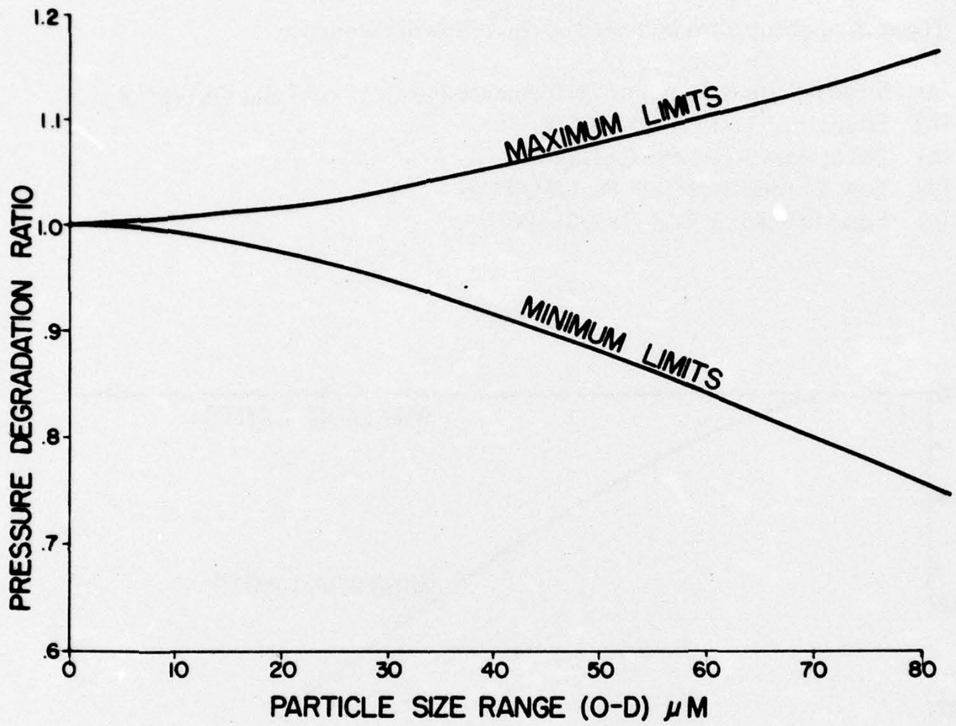


Fig. 2. Pressure Degradation Specification Limits.